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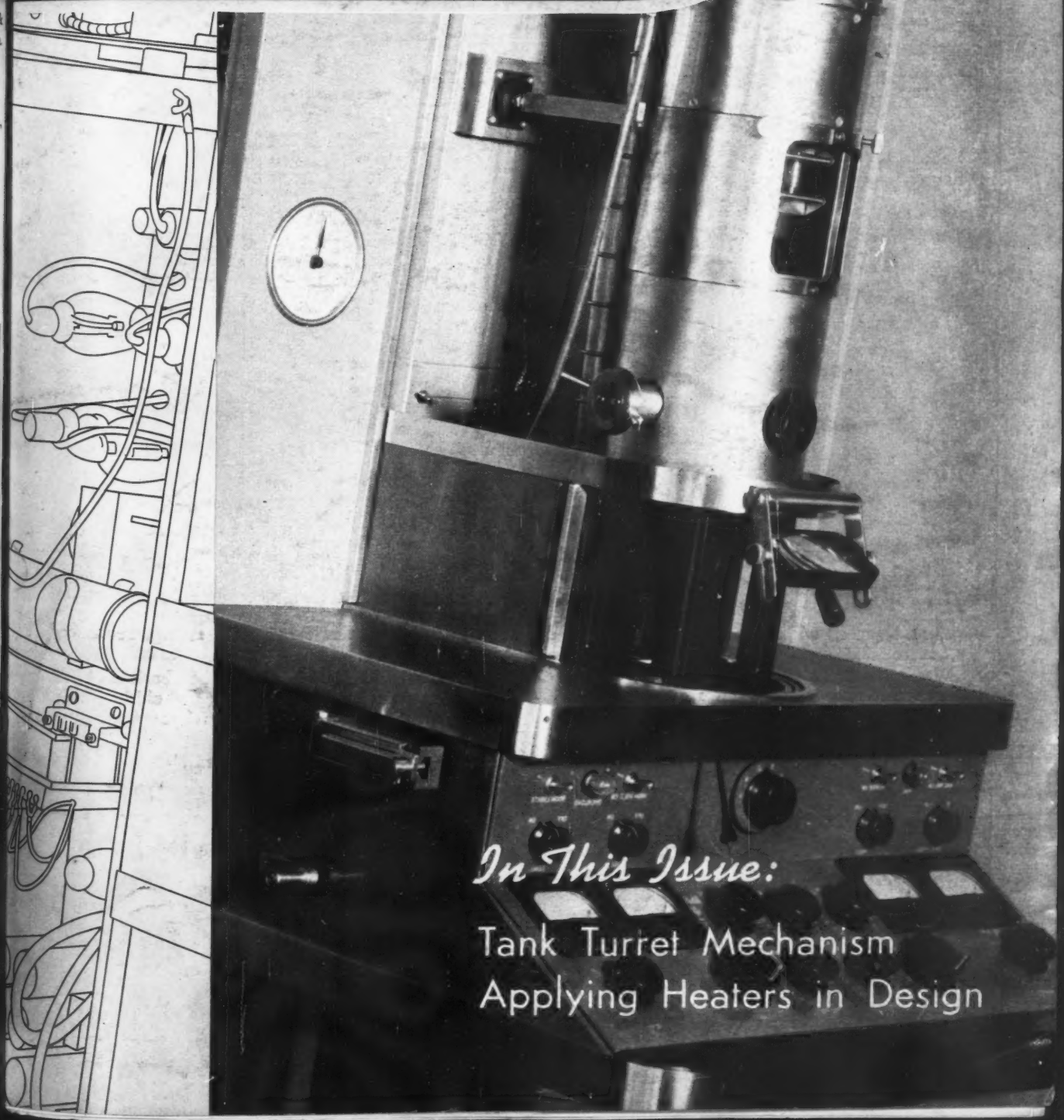
MACHINE DESIGN

March 1945



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Tank Turret Mechanism
Applying Heaters in Design

1 Pill for 9 Headaches!

PAGES FROM "A GUIDE TO WARTIME CARE OF ELECTRIC MOTORS"

Electric Motor Enemy No. 1...
DUST
DUST ON IT'S WINDINGS "BLANKETS" A MOTOR WITH A LAYER OF INSULATION.
Electric Motor Enemy No. 2...
STRAY OIL
OIL IS QUICK TO ATTACK INSULATION WHEN IT REACHES IT.
Electric Motor Enemy No. 3...
MOISTURE
HEAVEN for a motor most certainly would be a dry place. And while it is impractical to keep motors completely dry in many installations, that nevertheless should be one general aim of effective
How to Fight it...
Every effort should be made to keep water in liquid form from
Electric Motor Enemy No. 4...
FRICTION
IN THEORY, no wear can take place within a bearing if the
being moved and the floor under it.
But in actual practice, bearings
Electric Motor Enemy No. 5...
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ONCE MISALIGNMENT STARTS 'EM FIGHTING EACH OTHER...
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VIBRATION
VIBRATION SHAKES THE LIFE OUT OF MOTORS
Electric Motor Enemy No. 7...
UNEVEN WEAR
YOU EXPECT WEAR AT SOME POINTS...
(TIRES ON A CAR)
ON SLIP rings and commutators of electric motors, wear is accepted as inevitable—just as it is on the tires of a car. The reason, of course, is that none of these rotating parts exposed to constant friction can be fabricated with petroleum products.
But just as tire wear can be minimized by switching tires around to keep tread wear even—so the wear on commutators and slip rings can be minimized by preventing conditions leading to "grooving" (concentration of wear in narrow rings or cuts).
Electric Motor Enemy No. 8...
OVERLOAD
LIKE AN OVERLOADED HORSE, A MOTOR WORKED BEYOND IT'S HORSEPOWER CAPACITY IS SURE TO "BREAK DOWN".
TODAY, in a world at war
Electric Motor Enemy No. 9...
UNDERLOAD
ALTHOUGH the penalties are less severe, underloading an induction motor is just as clearly improper as overloading it.
As indicated below, the total current flowing in an a. c. circuit is made up of two components: (1) "real (useful) power"—that produces mechanical energy—and (2) "wattless current"—that produces merely the magnetic field required in induction motors.
It is characteristic of induction motors that when they are working under only part load the propor-
of wattless current is INCREASED.
Technically the "power factor" (ratio of real power to total power) is lowered.
Comparing total or apparent power to a man of a given size, we say that wattless current is like fat—real power like muscle. The size of one, the less of the other.
And because there is a limit to the power capacity of generators, transformers and power lines, the character or quality of the total power is of prime importance.

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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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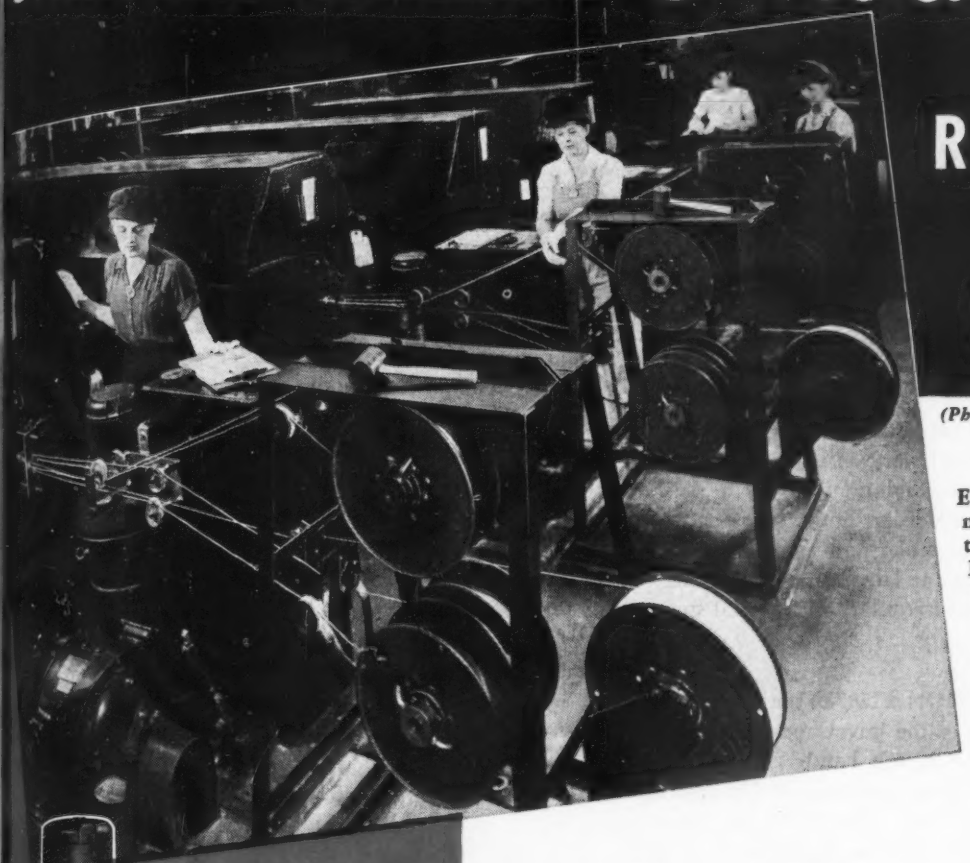

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RELIANCE A·C D·C MOTORS

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VARIETY OF COLORS are now allowed on metalworking machinery instead of being confined to "old machine-tool gray" as in a previous WPB order. The new order, however, permits only one coat of primer or sealer and not more than two coats of paint, enamel or lacquer. No filler may be applied to smooth out irregularities in metal surfaces.

STANDARDIZATION PLANS for the future insofar as joint action of the government and industry is concerned will be in the hands of a committee headed by C. E. Wilson, president of General Electric, acting in an advisory capacity to the American Standards Association and the Department of Commerce.

JUDGING BY ARTISTS' conceptions of postwar machines and products, one frequently gets the idea that modernistic motif and streamlining will be rampant in the future. While esthetic creations stir the imagination they often raise the question as to practicality. With bold design ideas at one extreme and limitations of materials, fabrication and maintenance at the other, in all probability the resulting compromise will be closer to orthodox than radical.

VOLUME OF WAR materials produced by General Motors alone involves more than 140,000,000 shells and casings; 180,000 cannon; 1,000,000 machine guns; 2,400,000 carbines; 180,000 airplane engines; 9000 complete bomber and fighter planes; 31,000 tanks, tank destroyers and armored cars; 740,000 trucks and a large percentage of all diesel engines. These figures for one manufacturer alone are indicative of the immense volume of equipment being produced.

COMPUTING AUTOMATICALLY and continuously the longitude and latitude position of a Superfortress, an air position indicator, developed by the Eclipse-Pioneer division of Bendix Aviation Corp., relieves the navigator of hours of tedious calculations. A single self-contained, lightweight device, it is mounted on the instrument panel and makes calculations which ordinarily would require charts, reference books, sextant, chronometer, parallel rules and dividers. The computer contains a system of built-in friction drives, balls and disks which delicately integrate information supplied by a remote indicating compass and a distance element. Obviating the chance of error through manual computation, the instrument has nearly 500 closely finished parts and eleven separate gear trains.

GERMANY IS CRAMPED for copper, nickel, molybdenum, vanadium, tungsten, chromium and manganese, in about that order, while Japan seems to have substantial stockpiles of alloying elements, according to a Battelle study presented in recent A.S.M. literature.

FIRST JET PROPULSION engine produced in this country from entirely original design is being developed at Westinghouse for the Navy. When completed the new engine is expected to produce more pounds of thrust for a given amount of fuel and pounds of weight than any engines brought from Europe for study.

AS FREQUENCIES for radio transmission increase it becomes more directional in effect. For instance it will be possible to transmit within a one-degree cone with a frequency of 2000 megacycles. Indicative of the trend in this direction, an application for license has already been made for as high as 12000 megacycles. Present plans for an installation of high-frequency transmission between New York and Boston will utilize a 2-watt booster antenna every 30 miles.

PATENTS COVERING electronic vulcanization of rubber originally granted to R. A. Dufour and H. A. Leduc of France and E. E. W. Dassner of Switzerland have been purchased by the B. F. Goodrich Co. and the Firestone Tire & Rubber Co. The patents will be made available to the rubber and plastics industries in the belief that ultimate developments will provide improved products at lower costs.

MACHINE DESIGN

Designing the Drive for the M-3 Turret

By Robert E. Becker
Chief Engineer
Logansport Machine Co. Inc.

Fig. 1—Turret at top of tank employs hydraulics for power and control

LIKE most hydraulically powered and controlled mechanisms, the heart of the turret-traversing mechanism of the M-3 medium tank is found in its valves. These, in the case of the M-3, have been combined into a single, compact unit that is of particular interest to the design field.

Before going into a detailed discussion of this "heart" unit, it might be

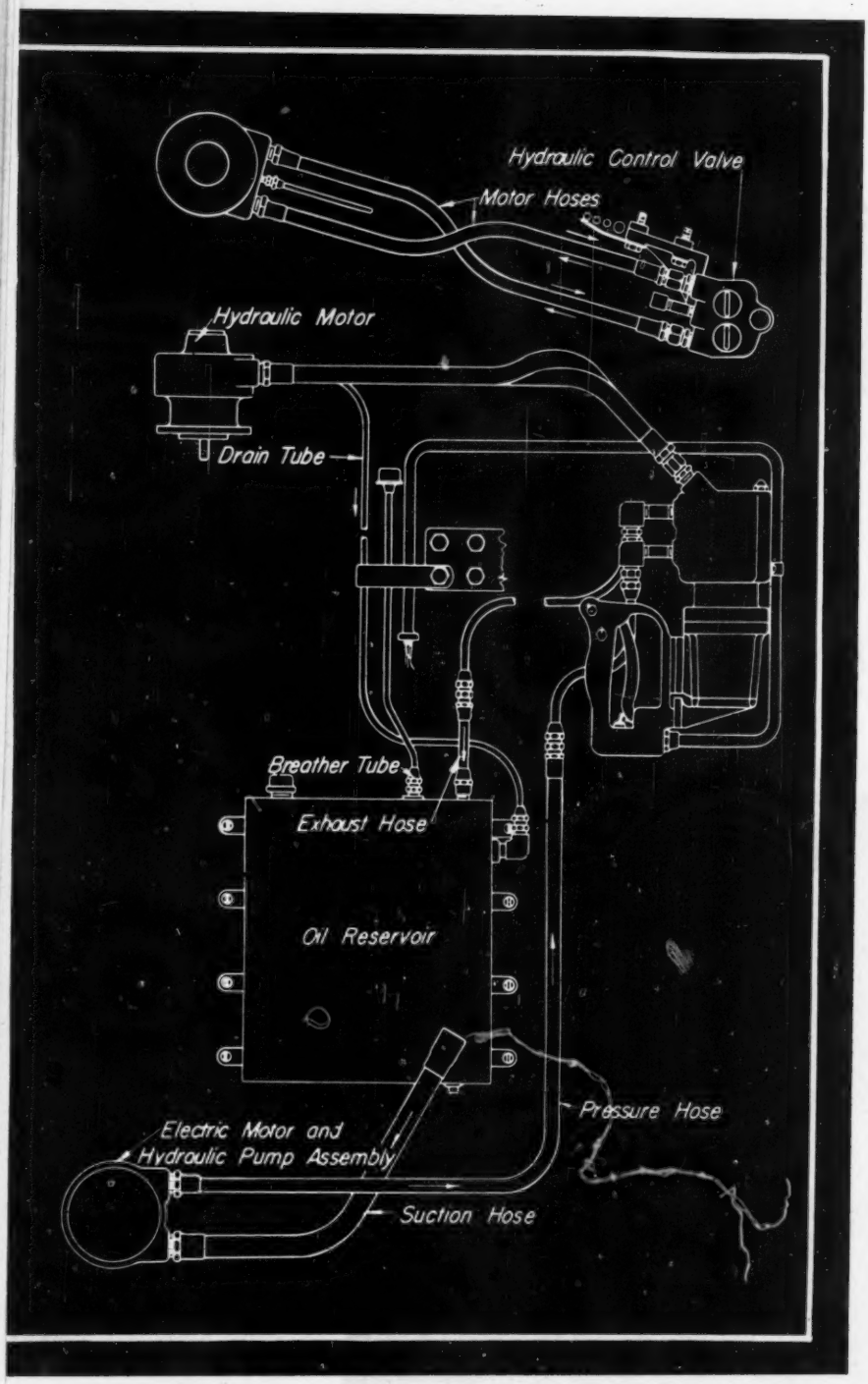


Fig. 2—Schematic diagram of hydraulic traversing system. Pump, motors, tubing, hose and fittings are all standard components

well to consider the entire turret mechanism so that a clear picture of the design problems involved may be obtained.

The job of the traversing mechanism is to rotate a heavily armored turret located at the top of the tank. This turret, as will be seen in *Fig.* 1, mounts three guns: Two 50-caliber machine guns and one 37-millimeter gun. In addition to the armor casing and the guns, the turret also carries a basket-like unit which fits down into a pocket or well in the tank proper, accommodating a gunner who, of course, turns with the turret.

Since the gunner is inside the turret, it is apparent that the controls must be there too. And while the power unit for driving the turret conceivably might be located outside the turret, multiple rotating joints with attendant sealing problems—to say nothing of awkward assembly and servicing—make outside mounting impracticable. Keeping all of the power and control units within the confines of the turret makes for simplicity in design and ease of assembly and maintenance.

What Requirements Demanded

During the preliminary design stages in the development of the M-3, definite requirements were laid down stating just what the turret should and should not do. These in turn demanded that the following requirements be met in the design of the traversing mechanism:

1. Turrets are notorious space misers, thus the drive and control must be compact. Further, it must be sturdy enough to withstand the abuse which normal handling in tank work entails
2. Units must be designed so as to lend themselves to quick assembly into the turret and convenient maintenance after installation
3. Right hand of gunner must be free for work other than turret control. This means the control unit must be designed for manipulation by the left hand
4. Acceleration and deceleration rate of turret traverse must be smooth and cover a range from zero to four revolutions per minute. Also, each time the control handle is turned to a given setting the traversing speed must be the same
5. Turret must not "creep" when locked by control handle at any setting. Therefore, once the gunner has set the turret pointing in any direction, neither standing the tank on a steep slope nor driving it over rough terrain must jog the turret from its setting
6. In the event the gunner is put out of action the turret should automatically lock in position. Also, provision must be made in the hand control to cut out the guns in this same eventuality.

One of the primary "musts" in working on the traversing mechanism design was to use as many standard units and parts as possible. There is sound logic behind this. Special parts are difficult and often impossible to procure in combat areas where minor repairs sometimes must be made under fire. Standard components are carried in general stock. This is particularly true of such items as screws, washers, nuts and bolts. Incidentally, utilization of standard components saved valuable design time which otherwise would have had to be expended in developing similar special parts and units.

A look at the schematic diagram of the traversing mechanism hydraulic system, *Fig.* 2, will show how closely this policy of using standard components was followed. The hydraulic pump, its driving motor, the hydraulic motor, and all fittings, hose and tubing, are

tandard commercial items. Only the control valve and il reservoir had to be designed specially to fit the job.

Briefly, operation of the mechanism consists of pumping oil from the reservoir through the pressure hose to the control valve, through the valve and one of the motor oses to energize the motor, out of the motor and back hrough the control valve from which it is led back to the eservoir through the exhaust hose. This, in essence, constitutes the hydraulic circuit. Add to this the gearbox (see *Fig.* 3) and the ability of the control valve to direct nd meter the oil flow, and the picture becomes complete.

This brings into focus the unique but simple design hich effectively combines into one unit the valves necesary to direct and meter oil under pressure from the pump. Under the control of its single handle this unit must:

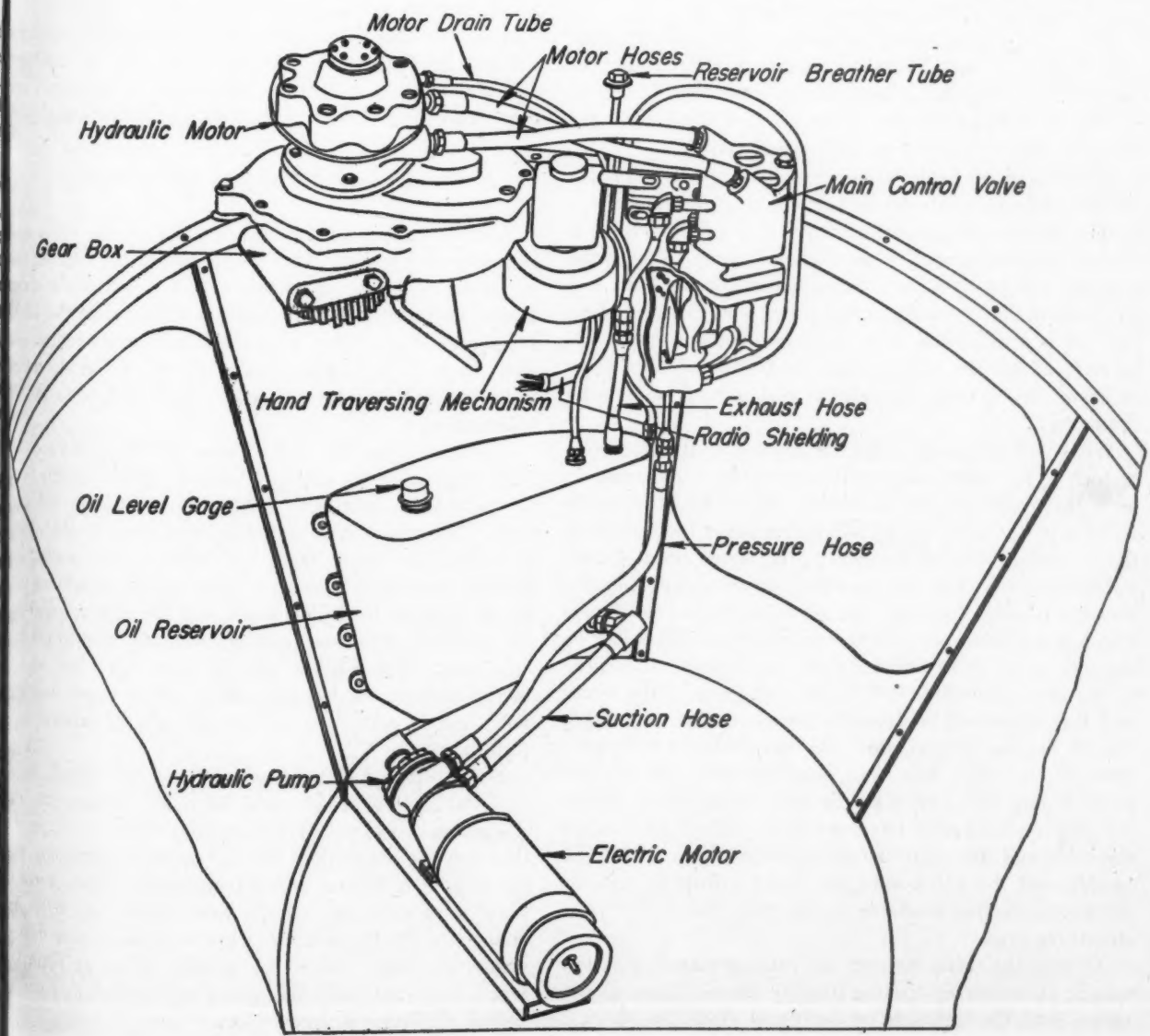
1. Depending on which direction the handle is turned, direct the pressurized oil through one or the other of the motor hoses in order to drive the motor in the desired direction
2. Meter the flow of oil, not only *to* the inlet of the motor, but also *from* its outlet.

The reason for directing the pressurized oil is self evident. However, the reasons for metering both the input and output of the hydraulic motor may not be quite as apparent. Thus, a few words of explanation are in order:

To enable the gunner to bring his guns into firing position quickly and accurately, it is necessary that the turret be capable of traversing at varying rates of speed, the speed being dependent on the amount by which the control handle is turned. To bring his guns into approximate position quickly, the gunner turns the handle hard over in the desired direction and the turret turns in that direction at its full speed of four revolutions per minute. As the turret approaches the desired position, the gunner eases the control handle back to its central or neutral position and the turret gradually—not abruptly—comes to a stop. Final adjustment of position then is effected by turning the handle slightly in the required direction.

Since a constant-delivery pump is employed, the variable speed demanded of the turret makes it necessary to meter the input and exhaust oil of the motor. The bypassing principle involved may be comprehended clearly

Fig. 3—Three-dimensional view of traversing mechanism installed in turret. Gear box contains reduction gearing and clutch to engage and disengage hand traversing mechanism which is used in case of power break-down



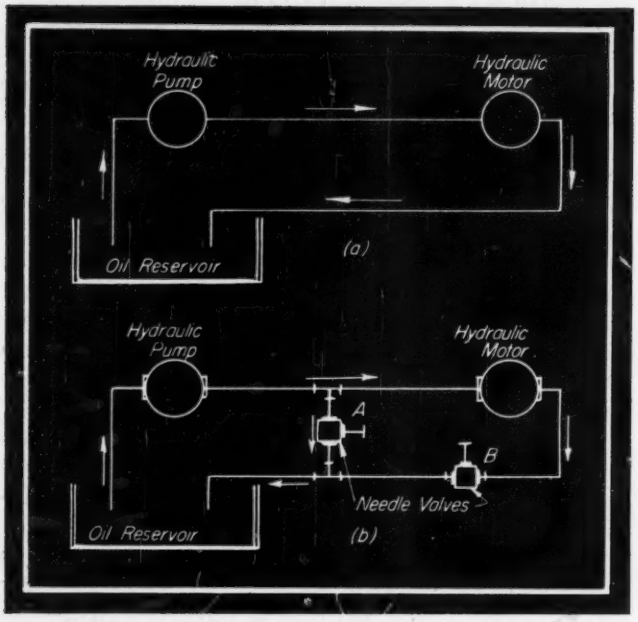


Fig. 4—Circuit a offers no means to control motor speed. Needle valves in b meter input and exhaust of the motor

by referring to the diagrams *a* and *b* of *Fig.* 4.

With only pipe connections between the reservoir, constant-delivery pump, hydraulic motor, and the return to the reservoir (*Fig.* 4*a*), the hydraulic motor would run at full speed in direct relation to the amount of oil delivered by the pump. This system lacks speed control.

Insertion of a needle valve, *A*, between the motor input and exhaust, as shown in *Fig.* 4*b*, changes the characteristics of the system considerably. If the needle valve is opened a small amount, some of the oil going through the pipeline will be by-passed through the valve. With less oil reaching the hydraulic motor, it will slow down. When valve *A* is opened still more, the speed of the motor will be reduced further. This could be continued until most or all of the oil goes through valve *A* instead of through the motor.

What will happen if a load is applied to the hydraulic motor? The motor then will require more pressure in order to overcome this resistance and, with the needle valve *A* partly open, the oil will find it easier to go through the valve than through the motor. However, because only a given quantity of oil can pass through an orifice when the pressure remains constant, the oil will endeavor to by-pass through the valve but will find that only part of it can go through unless more pressure forces it through. Thus, the resistance of the valve will become as great as the motor and the motor will continue to turn but with more force due to the rise in pressure. Oil will continue to by-pass through the valve but in a quantity controlled by the pressure imposed upon it by the load on the motor. When the load on the motor increases, more oil will go through the valve and this slows the motor down. If speed is to be maintained, the valve must be closed a little to raise its resistance, making available to the motor the higher pressure it requires.

To start the turret turning and running smoothly at low speeds (overcoming starting friction, the resistance of stiff grease and the tightness of the turret rings), a pressure greater than that required at normal running speeds is needed. This high starting pressure can be obtained by placing another needle valve (*B* in the diagram shown in *Fig.* 4*b*) in the exhaust line of the hydraulic motor. This will cause more oil to be by-passed through valve *A* because valve *B* restricts the flow through the hydraulic motor, giving rise to an increase in pressure. Very little oil is necessary at low speeds but the pressure must be sufficiently high. The restricting valve *B* must be opened when the speed of the hydraulic motor is to be increased. This brings the pressure down to that required for normal running speeds.

Referring now to the M-3 turret control valve shown cross-sectionally in *Fig.* 5, it will be seen that the pistons and liners employed comprise a combination of the needle valves *A* and *B* mentioned in the foregoing. Two pistons and two liners are used, one pair for directing the oil flow to drive the motor clockwise, and the other pair for directing the oil to drive the motor counterclockwise. To actuate the pistons under the control of the single handle a rack and pinion drive is utilized, the racks being fastened to the lower ends of the pistons while the pinion is keyed to the shaft of the handle.

It is interesting to note, in passing, the manner in which the racks are attached to the pistons. Links, similar to those used on roller chain are employed. This construction was necessary because of the commercial impracticability of machining the rack seats in the lower casting in accurate alignment with the liner bores in the upper casting. The links make the pistons and racks self-aligning.

Valve Automatically Self-Centers

Another purely mechanical feature of the valve is its self-centering action. This is accomplished by spring loading each piston in compression and providing a spring loaded detent pin in the handle. Thus, when the handle is released, the springs drive the handle to its central position where it is locked by the action of the detent pin snapping into engagement with a bushed hole in the bottom casting.

In each of the two valve liners are three rows of four holes spaced at 90 degrees to each other. Each of the upper holes is vertically located over those in the lower row. The upper end of the liner is open to the exhaust core and the upper holes are open to the motor line through the motor line core. The middle holes are offset by 45 degrees from the upper and lower holes and both the middle and bottom rows of holes are open to the pressure core. The middle holes in each liner line up with the by-pass grooves in each piston and the upper and lower holes line up with the pressure and exhaust grooves in the pistons.

Each piston has four exhaust grooves which engage with the top row of holes and the motor oil line core when the piston descends. The exhaust grooves are tapered at their lower ends so they will act as restrictions only when the handle is turned a slight amount. (They serve the same purpose as the needle valve *B* of *Fig.* 4*b*). The connection of the motor line to the exhaust core does not occur, of course, when the handle is set at neutral, at which time the hydraulic motor is blocked and will not permit the turret to move.

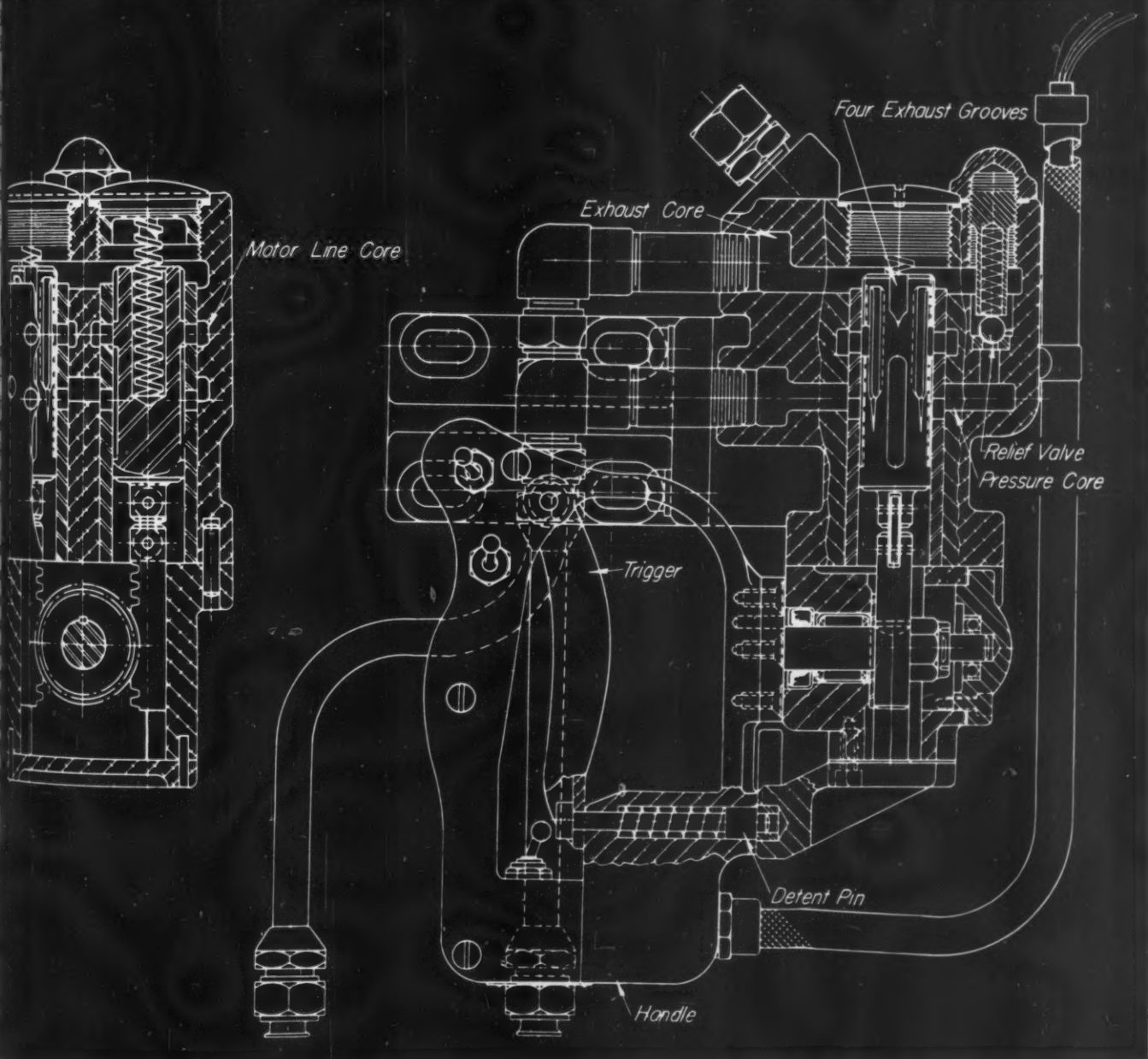
n line with, but below the exhaust grooves in the pis-
s, are pressure grooves which connect the lower row
liner holes with the top row when the piston ascends.
ey are blocked off in neutral and also when the piston
cends.

At 45 degrees to the exhaust grooves in the piston are by-pass grooves. In neutral, these connect the middle of liner holes and the pressure core to the top end of liner and the exhaust core. The lower ends of these pass grooves are tapered to a sharp point and act as ices when the piston ascends. The farther the piston es, the less is the area of the grooves exposed to the dle row of holes until the two are disconnected en-ly. These by-pass grooves serve the same function as needle valve *A* in the circuit of *Fig.* 4*b*.

When the handle is in neutral position the two pistons are of course at the same level. The exhaust grooves and pressure grooves are blocked off and in consequence the motor is blocked also. At this time oil entering the pressure core passes through the by-pass grooves in the pistons to the exhaust core at the top end of the liners.

When the handle is turned clockwise the piston on the left, *Fig.* 5, rises, while the piston on the right lowers. The pressure grooves of the ascending piston connect the pressure port to the motor line. The by-pass grooves (needle valve *A* of *Fig.* 4*b*) are still open so the quantity of oil going to the motor is small and, in consequence, the motor will turn at minimum speed. If the handle is turned

Fig. 5—Cross section of main control valve shows how single-handle control actuates pistons that direct and meter oil flow to and from the hydraulic motor



further in the clockwise direction—say 30 degrees—the pressure grooves still will connect the pressure port to the motor line. However, the by-pass grooves will have a smaller area exposed to the middle row of holes due to the taper of the grooves. This would be the same as partially closing the needle valve *A* in the diagram of *Fig. 4b*. The more the handle is turned the less is the area exposed to the middle row of holes through the by-pass grooves. Consequently, more and more of the oil must go through the pressure grooves to the hydraulic motor line, causing the speed of the motor to increase. When the handle is turned to the wide open position, the by-pass grooves (needle valve *A*) are closed completely and all of the oil must go through the pressure grooves to the hydraulic motor.

This explains what happens when one of the pistons rises. However, the action of the valve depends also on what occurs due to the simultaneous lowering of the other piston. The exhaust grooves of the descending piston (needle valve *B*) connect, via the motor line core, to the exhaust port. The exhaust grooves are tapered to a p where they engage the top holes (motor ports) of liner. Therefore, as the piston descends, at first very then more and more groove area will be exposed to by- exhaust oil from the motor. Meanwhile, the by- grooves are closed off and the pressure grooves re blocked.

If the handle were turned counterclockwise, the p on the left would go down and the one on the right w go up. This would cause the hydraulic motor to tur the opposite direction because the right piston would directing oil into the motor and the left piston would metering oil from the motor to the reservoir.

As has been mentioned in the foregoing, the handle the main control valve had to be designed for manip tion by the left hand. Reference to either *Fig.* 5 or Fig will show three switches located on the handle. One, the base of the handle, is actuated by the trigger mec and the other two are located so they can be snapped o veniently by the thumb of the left hand. These la two control the fire of the turret guns, while the forme a cutout, which, with the trigger in its normal position shown in *Fig.* 6, isolates the gun firing circuits from two upper switches. Thus, when the gunner wishes steer the turret and fire its guns, he must first squeeze control handle trigger. This action unlocks the handle steering and also, by throwing the switch at the bas the handle, completes the firing circuit to both u switches. Should something happen to take the gu out of action, relinquishment of his grip on the handle release the trigger. The freed handle then will return its neutral position automatically and at the same time release of the lower switch will cut out the firing circ This system makes impossible the unguided, indiscrimi fire which would occur if no automatic cutout or self- tering handle were employed.

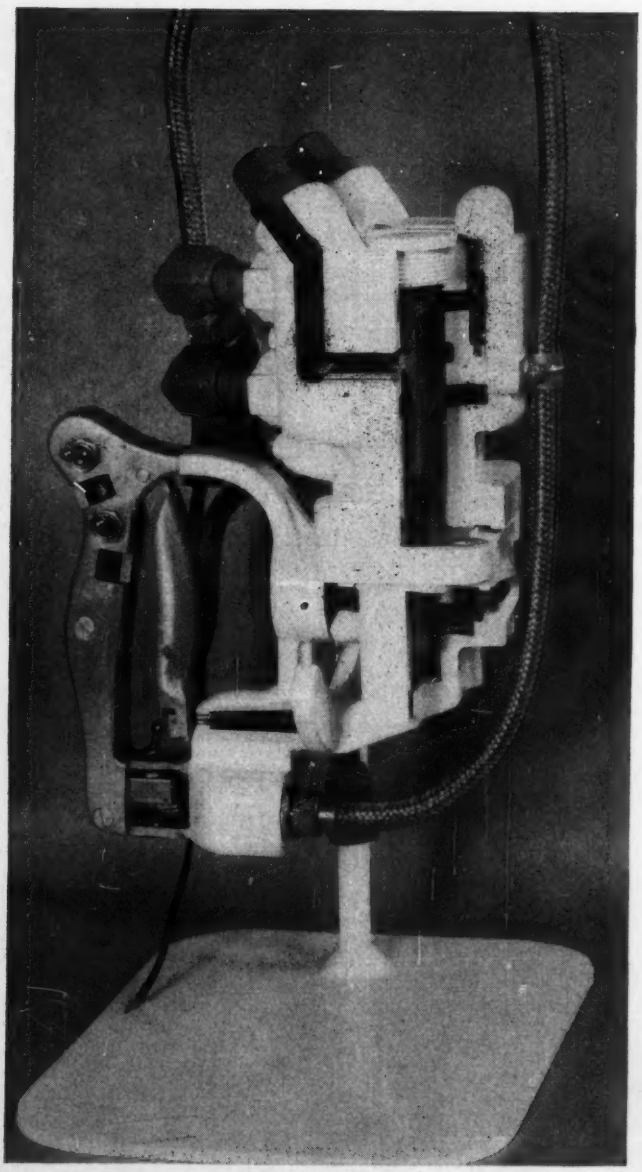


Fig. 6—Control valve is physically cross-sectioned to show primary operating parts. Spring-loaded detent pin at base of trigger locks handle in neutral position

Advantages Offered by Low-Pressure System

During the initial design stages of the M-3 hydraulic cuit, it was tentatively planned to provide a system p sure of 1000 pounds per square inch. However, this l was lowered to 600 pounds per square inch because higher pressure entailed not only the ever-present da of bursting lines but, in addition, would cause exces drain on the batteries employed to supply the 24-volt rect-current pump motor.

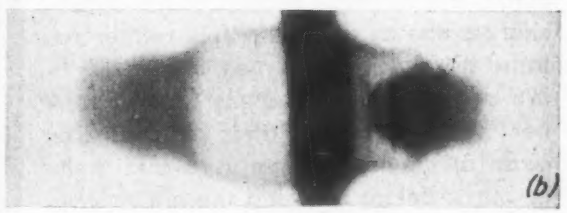
Since the M-3 tanks were to be used throughout gamut of the temperature range embraced by Arctic tropic combat areas, it was necessary to specify a hydra oil that would operate satisfactorily from −65 to 140 grees Fahr. While military security does not permi vulgence of the identity of this fluid, suffice it to say the specifications were satisfactorily met.

It was during the hectic days just after Pearl H that the M-3 traversing mechanism literally sprang the drawing boards at Logansport to meet the "on double" demands of our army's sudden venture into out war. However, even though this mechanism has strictly a wartime development, the design principle volved—particularly in the compact control valve well point the way to more proficient utilization of draulics in the ever better machines of the future.

Scanning THE FIELD for Ideas

High-speed X-ray pictures can now be used to dy the action of component parts of machines even ugh their action be obscured from normal vision by closures or other machine parts. Indicative of the pos- ilities for studying valve action and other mechanisms der operating conditions is the series of high-speed iographs, right, made at the Army's Frankfort Arsenal. owing progressive penetration of a high-explosive ll through a steel plate, the nose of the shell is just sting through the plate at (a), almost half the shell has sed through and swelled to approximately twice its rmal diameter at (b), at (c) the shell has just burst en, while at (d) the shell has burst wide open and the als on the back of the plate are curling to make a ge hole. Other tests showed the passage of a bullet ough the gun barrel after firing, indicating the align- nt of the bullet as it zips along the gun's bore. Causes

(a)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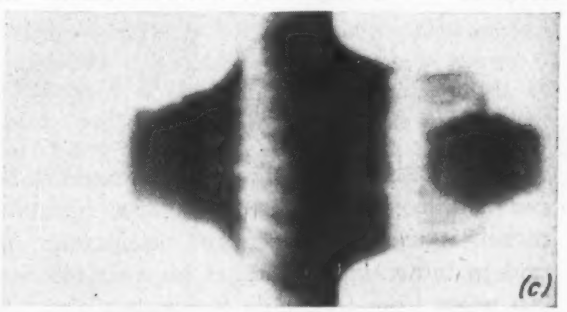
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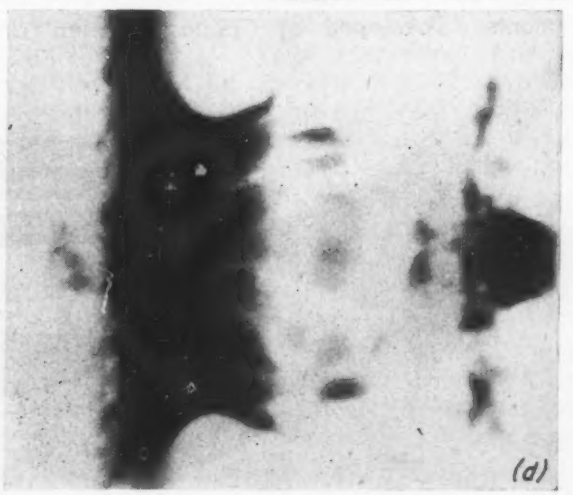
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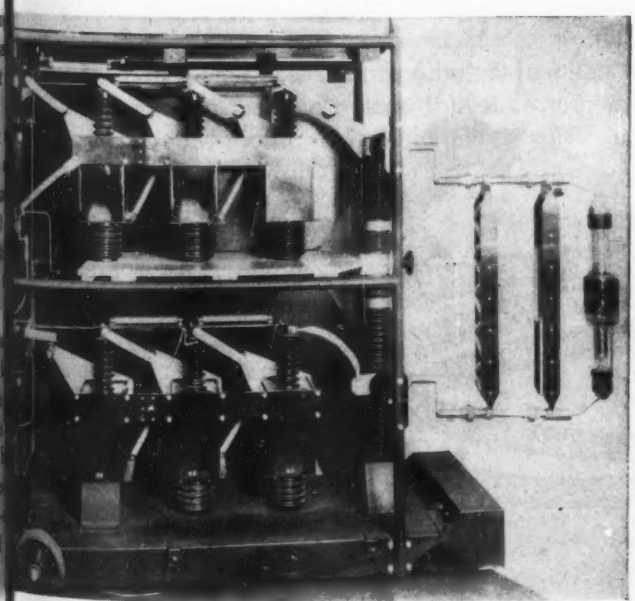
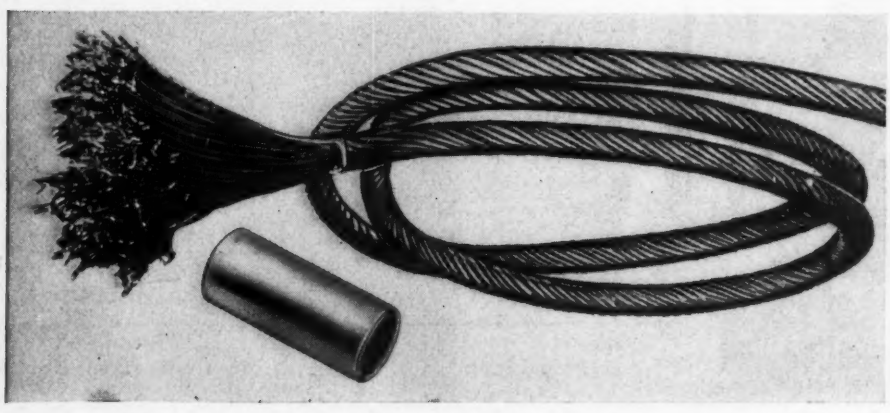


for jamming and other faulty action may be studied in this way.

Also illustrated, is the surge generator for producing the sudden burst of 600,000 kilowatts of electricity to make exposures of a millionth of a second required for these studies. The X-ray tube, developed by Westinghouse Lamp division, is especially designed to jerk the electron out of a cold metal cathode at an unusual rate of speed when a high momentary voltage is applied. A vacuum metallic arc is produced, providing a path across which the electrons race until they strike the anode, which in turn results in pouring a barrage of X-rays through solid metal to make the photographs.

Gaskets of silicone rubber, that versatile compound, are being utilized on turbosuperchargers for the B-29 Superfortress. An excellent material for high-temperature sealing, its flexibility is indicated by the curve it has assumed across the cutaway section of the supercharger shown at right. This rubber retains its elastic properties from −60 to 575 degrees Fahr. whereas natural rubber becomes brittle and smashes like glass at the low temperature and melts or burns before reaching the higher one.

Composite wire, having a core of 70 per cent copper clad with 30 per cent nickel, is useful in electronic equipment and in industrial apparatus for connecting wires from terminals to heating elements. Developed by Callite Tungsten Corp., the wire shown below has good conductivity and resistance of nickel to oxidation and corrosion. Known as grid, the wire is made by a process which draws down components of the wire at the same time. A billet from which strands are reduced is shown alongside the cable in the illustration. This cylinder section of bimetal rod may be drawn in a wide range of sizes including the .006-inch fine of the flexible stranded wire shown. Such wire can be subjected to cherry-red heats without losing its flexibility. It resists fatigue and embrittlement and withstands stresses many times greater than copper will without breaking.



Worm-Drive "Jitters" Can Be Avoided

By S. J. Mikina
Westinghouse Research Laboratories

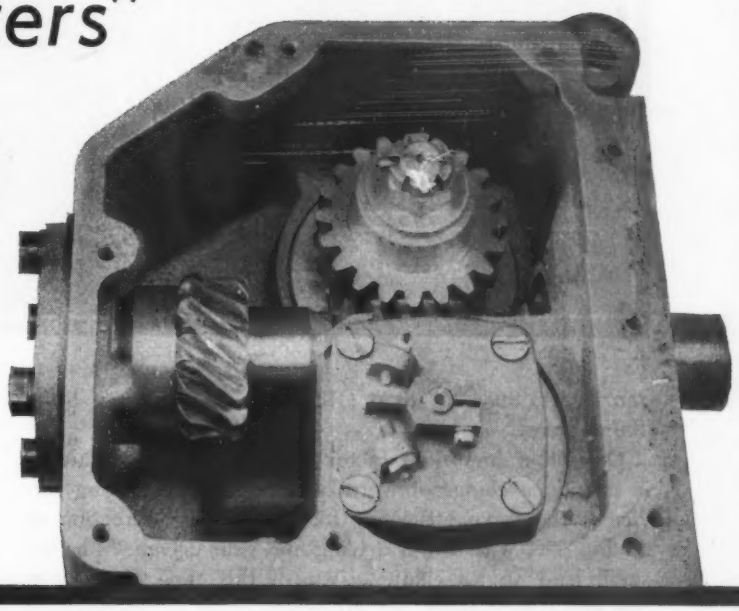


Fig. 1—This worm drive for a 6-ton gun turret was subject to jerky operation under certain conditions. Successful cure resulted from analysis in accompanying article

WHEN irreversible, or self-locking, worm gearing is required to move a load that tends to drive the worm and its connected power source, as in the e of a crane lowering a load or during depression of a avity-unbalanced antiaircraft gun, difficulty may arise e to uneven operation. Under these conditions a cer- definite relationship must exist between the effective ss inertia of the worm about its bearing axis and the ss inertia of the overhauling load if smooth, chatter- s operation is to be realized. When this inertia rela- nship does not exist, intermittent separation of the rm and gear teeth will occur at the point of irreversi- ity and result in jerky operation because, alternately, e worm advances ahead of the gear through the gear earance and then the descending load causes the gear catch up with the worm and to engage it with impact. is jerky condition is particularly pronounced at low or eping speeds of the drive, when insufficient velocity sts at the gears to maintain oil film lubrication of the ting surfaces. The British have aptly named this enomenon "stairstepping" because the spasmodic pro- ession of the load under these conditions usually oc- s with a fairly regular period of ½ to 1 second and is s suggestive of the bumpy descent of a stairway.

It is the object of this article to present a simple mathe- tical analysis of the phenomenon of stair-stepping of rm gearing and, on the basis evolved, to point out a ctical means for its elimination.

The condition for irreversibility of worm gearing is pressed by the well-known criterion, which requires t

$$\cos\phi \tan\lambda \leqq \mu \quad \ldots\ldots (1)$$

which ϕ = pressure angle of worm profile, λ = lead angle of worm and μ = coefficient of friction at region of contact between worm and gear. The pressure angle ϕ is well defined by the worm geometry, varying from 0 degrees for a square-threaded helix to 14½, 20 or 30 degrees for commercial worm gearing, and 30 degrees for the standard V-thread screw. The coefficient of friction μ, however, is a highly variable quantity in worm gearing, depending as it does on such factors as gearing materials, accuracy of profiles, kind of surface finish, kind of lubricant and manner of feed, as well as on the relative sliding velocity at the contacting surfaces, on the applied gear load, and on the amount of mechanical vibration transmitted to the worm and gear contacting surfaces.

Irreversibility a Function of Friction Coefficient

Irreversibility of a given set of gears thus is not definitely assured over its entire range of operation. A set of worm gearing may be irreversible at standstill, when the coefficient μ is large, but may become reversible when an oil film develops at the contact surfaces as sufficient sliding velocity is acquired.

For the usual adequately machined well-lubricated gearing consisting of a hardened and ground steel worm and a hard bronze alloy gear, the principal variable affecting the coefficient of friction is the rubbing velocity of the mating surfaces. Then μ may vary from as high as .15 at standstill to as low as .01 at rubbing speeds in the neighborhood of 4000 feet per minute*. For the 14½-degree pressure angle worm gearing, for example, the limiting worm lead angle λ for irreversibility may vary from

*H. E. Merritt—*Gears*, Page 243, Isaac Pitman & Sons Ltd.

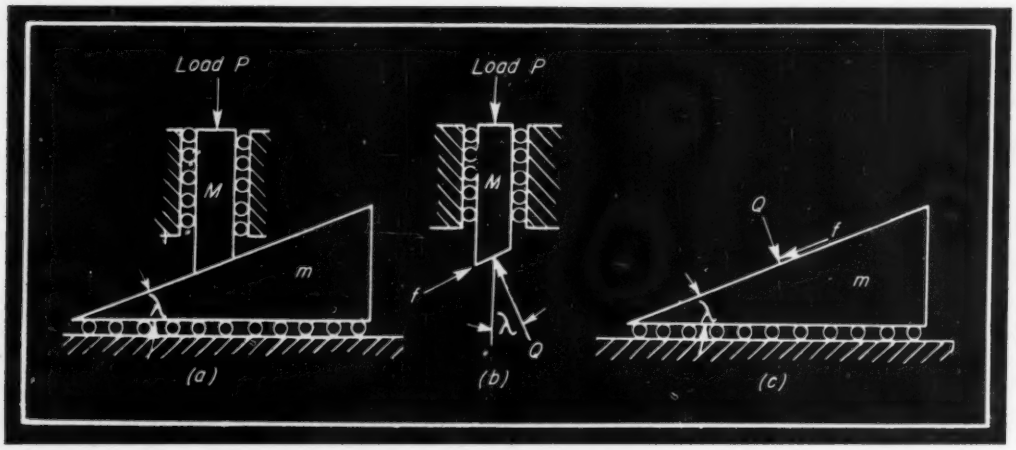


Fig. 2 — Left — Wedge mechanism, which forms basis of a worm drive with zero pressure angle, is shown in condition of self-locking equilibrium

$tan^{-1}(.01/.968)$ to $tan^{-1}(.15/.968)$, that is, from .59 to 8.9 degrees.

One other important factor which must be considered in making a choice of worm gearing for self-locking service is the drive efficiency for the forward direction of motion, i.e., with worm driving and gear driven. For the small lead angles characteristic of this type of gearing, the usual expression for efficiency may be reduced to

$$e=100\frac{\lambda \cos \phi}{\mu+\lambda \cos \phi} \quad \ldots\ldots (2)$$

in which the lead angle λ is now expressed in radians.

The author first encountered stair-stepping trouble in a gun-turret drive of his design, *Fig.* 1, in which it was desired to secure a fairly efficient speed reduction of 20:1 in the smallest possible space and to obtain irreversibility with the worm at slow creeping speeds or at standstill. The requirement of good forward efficiency at higher operating speeds is not incompatible with the retention of irreversibility at low speeds and particularly at standstill. In this case, a worm with a lead angle of 4⅔ degrees was chosen for a duty involving a maximum rubbing speed of approximately 500 feet per minute. The coefficient of sliding friction at that speed was about .02, hence the gearing efficiency at the maximum speed was, from Equation 2:

$$e=100\frac{\frac{4.66}{57.3}\times.968}{.02+\frac{4.66}{57.3}\times.968}=80 \text{ per cent}$$

At low or creeping speeds, the sliding velocity was insufficient to maintain a separating oil film between worm and gear and μ rose to .12, thus causing the criterion, Equation 1, to be satisfied by a sufficient margin to insure consistent irreversibility in the neighborhood of standstill.

Performance of the drive was up to expectations in all respects but one: The gun turret in question was at times subject to gravity unbalance, and the worm drive exhibited a fine case of jittery stairstepping whenever the turret had to be driven slowly in the direction in which the gravity unbalance was urging it to go. What made the problem particularly baffling were the facts that the gu turret could be driven "uphill" against the gravity u balance without the least bit of chatter at any spe down to standstill, and could be driven "downhill" wi out stairstepping at all speeds above a certain minim speed equal to about one-tenth of the maximum requir operating speed.

To explain the observed facts of the case, the follo ing analytical approach to the problem was made. Co sider first the inclined plane of *Fig.* 2 when it is used a wedge for moving a mass up or down. This incli plane is the basic form of a worm drive of zero pressu angle (a developed square-threaded screw) and shou exhibit all of the important mechanical characteristics worm gearing, including that of stair-stepping. Let t inclined plane of mass m be s[illegible]orted by smooth rol of negligible friction (analogous to ball bearings on t worm shaft), the engaging mass M being similarly co strained by roller guides of negligible friction for verti motions of M (analogous to ball bearings on the wor gear). The point of contact of the mass M with the i clined plane has a coefficient of friction μ. Moreov since the analysis concerns an irreversible mechanism, t criterion of Equation 1 must be satisfied: Angle λ of t inclined plane must be such that $tan\lambda \leqslant \mu$. Finally, let t mass M be subject to a total downward force P pressing against the inclined plane.

At standstill, the equilibrium of the system of Fig requires no additional external forces other than the re tions of the frictionless rollers. The applied force P mass M gives rise to a normal reaction Q at the incli plane and a fractional force f along the inclined pla This frictional force is just sufficient to balance the co ponent Q tending to push the inclined plane to the ri and away from M. Thus the horizontal equilibrium of t inclined plane requires that

$$f \cos\lambda = Q \sin\lambda \quad \ldots\ldots$$

as pictured in *Fig.* 2*c*.
Vertical equilibrium of the mass M requires that

$$Q \cos\lambda + f \sin\lambda = P \quad \ldots\ldots$$

as shown in *Fig.* 2*b*. These equations give

$$Q=\frac{P}{\tan\lambda\sin\lambda+\cos\lambda} \quad \ldots\ldots (5)$$

d

$$f=Q\tan\lambda=\frac{P\tan\lambda}{\tan\lambda\sin\lambda+\cos\lambda} \quad \ldots\ldots (6)$$

ce in this case $\tan\lambda \leqslant \mu$, the value of f calculated from uation 6 may be less than the maximum developable ctional force μQ.

In order to cause the system of *Fig.* 2 to move, a horiontal force F must be applied to the inclined plane in the rection of the desired motion. Uphill motion of the ass M and its associated load P exhibits no anomalies. that case, as the force F is gradually increased in magtude, but is still less than that required to initiate motion the system, standstill is still enforced, but the self-locked ndstill equilibrium Equations 3, 4, 5 and 6 are disbed to the extent that both the normal force Q and the ctional force f on the inclined plane also change in magtude. Ultimately f even goes through zero to a reversed rection. As f reverses in direction and finally becomes ual to μQ, motion of the system is started by the ocrrence of slipping at the point of contact between M d m.

No Separation Tendency for Uphill Motion

Subsequent acceleration of both m and M due to a furer increase in F or to a reduction in the coefficient of ction from its static value to the dynamic or sliding coficient ν will serve to increase or to maintain the contact ssure between m and M, since M can only be accelated uphill by its reaction forces from the inclined plane The starting transient for the upward motion of M us shows no tendencies for a reduction in the contact rections and a possible consequent reduction to zero folwed by contact separation. Occurrence of such contact paration is an obviously necessary preliminary to the velopment of chattering or stair-stepping. The smooth gression of the mass M and its load P for uphill motion a familiar experimental fact.

The situation is found to be radically different, however, on investigating the consequences of attempting to move M downhill in the direction of the applied load P by pulling on the inclined plane to the right, as shown in *Fig.* 3. At the start, i.e., at self-locked standstill, the frictional force f is less than μQ. As the tractive force F is increased in magnitude, f increases also until it reaches a limiting value equal to μQ, when slippage starts between m and M. value equal to μQ, when slippage starts between the inclined plane m and the engaging mass M.

Equilibrium relations just at the inception of motion, when F is an infinitesimal amount greater than the value required to make $f=\mu Q$, are shown in *Fig.* 3. Horizontal equilibrium of the inclined plane now requires that

$$F=\mu Q\cos\lambda-Q\sin\lambda \quad \ldots\ldots (7)$$

Vertical equilibrium of the mass M is given by

$$P=\mu Q\sin\lambda+Q\cos\lambda \quad \ldots\ldots (8)$$

Solving for F,

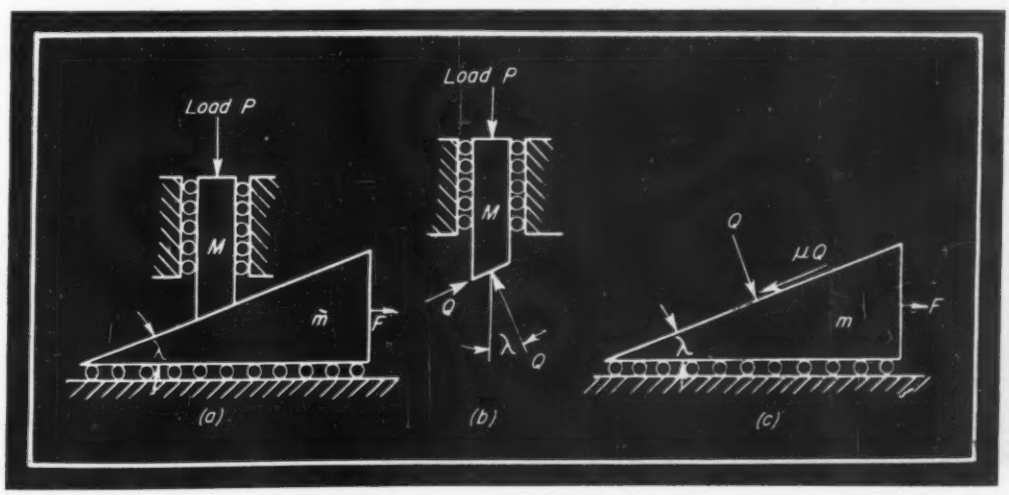
$$F=P\frac{\mu-\tan\lambda}{1+\mu\tan\lambda} \quad \ldots\ldots (9)$$

As soon as the tractive force F exceeds this value by an infinitesimal amount, the inclined plane will begin to move. Now, however, in contradistinction to the case of uphill motion, there is no assurance that M will follow m because, as the inclined plane starts to move, it can conceivably have a higher acceleration than the mass M can follow at the instant that the contact pressure between m and M drops to zero. Separation of the contact surfaces will then occur and will increase until either the force F is reduced in value or until other forces come into play, such as occur when gear clearances are taken up. In order to test the postulate of instability of contact pressure and to determine the conditions for separation of m and M, it is assumed that an infinitesimal separation of the just contacting surfaces has occurred and then the relations governing the increase (or decrease) of the postulated separation are written down.

Since the mass of the inclined plane is m, its initial acceleration will be, from Equation 9:

$$A_m=\frac{F}{m}=\frac{P}{m}\,\frac{\mu-\tan\lambda}{1+\mu\tan\lambda} \quad \ldots\ldots (10)$$

The contact surface of the inclined plane will draw away



g. 3—Right—Wedge echanism is represented in a condition equilibrium at the rt of "downhill" motion

from the mass M with a vertical acceleration

$$A_v = A_m \tan \lambda$$

or

$$A_v = \frac{P}{m} \frac{(\mu - \tan \lambda)\tan \lambda}{(1+\mu \tan \lambda)} \quad \ldots\ldots (11$$

Vertical acceleration of the follower mass M due to the load force P will be

$$A_M = \frac{P}{M}$$

Now, if contact separation between m and M is to be avoided, A_v must be equal to or, preferably, it must *tend* to be less than A_m. This results in the following criterion for no separation:

$$\frac{P}{m} \frac{(\mu - \tan \lambda)\tan \lambda}{(1+\mu \tan \lambda)} \leqq \frac{P}{M},$$

or

$$m \geqq \frac{M(\mu - \tan \lambda)\tan \lambda}{1+\mu \tan \lambda} \quad \ldots\ldots (12)$$

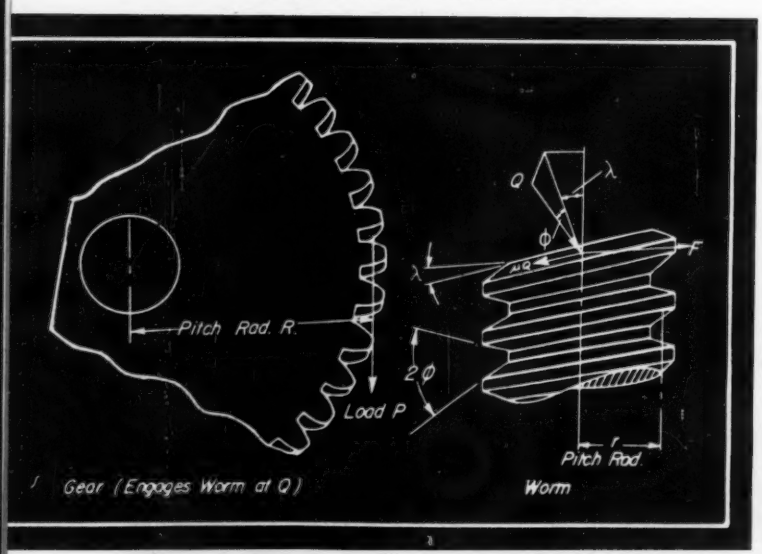
For small λ, which is the case for self-locking inclined planes, this criterion reduces to

$$m \geqq M\lambda(\mu - \lambda) \quad \ldots\ldots (13)$$

It is to be noted that the above criterion is independent of the load P, except insofar as the load affects the coefficient of friction μ.

The criterion, Equation 12, is just a precise statement of the more or less obvious fact that if m is made large enough then the force F, which bears a definite relationship to the load P at the inception of motion, can never accelerate the inclined plane so fast that the follower mass M (under the urging of load P) cannot follow it.

Fig. 4—Actual worm and gear are shown in equilibrium at the start of "downhill" motion



The foregoing analysis may readily be extended to worm gearing as follows: The inclined plane which has been considered thus far may be used to represent a developed segment of a worm, while the engaging member M pressed against it by the load P represents the worm gear. Such gearing has mating surfaces which, at the point of contact, are at an angle ϕ relative to the axis of the worm as shown in *Fig.* 4, the reaction Q is no longer in the vertical plane but has the angle ϕ with the plane of the paper. As before, the self-locked condition of worm and gear is first analyzed and then the equilibrium relations pertaining to downhill starting of the load investigated. Downhill in the general case refers to the motion of M in the direction of the gear load P.

Let the tangential force exerted by the external driving torque on the worm be represented by the force F acting on the worm pitch line as shown in *Fig.* 4. Then as F is increased until motion is started, the tangential equilibrium of the worm segment requires that at the inception of worm motion

$$F = \mu Q \cos \lambda - Q \cos \phi \sin \lambda \quad \ldots\ldots (14)$$

analogous to Equation 7 for the simple inclined plane. The vertical equilibrium of the engaging gear segment is given by:

$$P = \mu Q \sin \lambda + Q \cos \phi \cos \lambda \quad \ldots\ldots (15)$$

analogous to Equation 8 for the basic inclined plane. Solving for F,

$$F = P \frac{\mu - \cos \phi \tan \lambda}{\cos \phi + \mu \tan \lambda} \quad \ldots\ldots (16)$$

As before, it is assumed that a minute gear separation has occurred at the start of the motion; then the conditions for increasing or decreasing the postulated separation may be written down. If r is the pitch radius of the worm, the angular acceleration of the worm upon loss of contact with the gear is

$$\alpha_w = \frac{Fr}{I_w} \quad \ldots\ldots (17)$$

where α_w = worm acceleration, radians per second per second, and I_w = rotational inertia of worm and associated drive mechanism about the worm bearing axis. The tangential acceleration of the worm segment that was in contact with the worm gear is

$$A_{wt} = r\alpha_w = \frac{Fr^2}{I_w} \quad \ldots\ldots (18)$$

Acceleration with which the sloping worm segment is withdrawing vertically from the gear is

$$A_{vw} = A_{wt} \tan \lambda = \frac{Fr^2 \tan \lambda}{I_w} \quad \ldots\ldots (19)$$

Angular acceleration of the worm gear and its associated load P is

$$\alpha_p = \frac{PR}{I_p} \quad \ldots\ldots (20)$$

where α_p = acceleration in radians per second per second, R = pitch radius of worm gear and I_p = rotational inertia of worm gear and associated load P referred to worm gear bearing axis.

Vertical acceleration (i.e. along axis of worm) of the gear segment that was in contact with the worm is

$$A_{vp} = \alpha_p R = \frac{PR^2}{I_p} \quad \ldots\ldots (21)$$

The condition for avoidance of worm and gear tooth separation is then

$$A_{vw} \leq A_{vp} \quad \ldots\ldots (22)$$

$$\frac{Fr^2 \tan \lambda}{I_w} \leq \frac{PR^2}{I_p}$$

Now, the ratio of the worm angular velocity to the angular velocity of the worm gear is

$$N = \frac{R}{r \tan \lambda} \quad \ldots\ldots (23)$$

Substituting N and the value of F from Equation 16 in Equation 22, the criterion for smooth driving downhill is finally obtained:

$$I_w \geq \frac{I_p}{N^2} \frac{\mu - \cos \phi \tan \lambda}{\tan \lambda (\cos \phi + \mu \tan \lambda)} \quad \ldots\ldots (24)$$

For the small lead angles λ of irreversible gearing, Equation 24 simplifies to

$$I_w \geq \frac{I_p}{N^2} \frac{\mu - \lambda \cos \phi}{\lambda \cos \phi} \quad \ldots\ldots (25)$$

The Reasoning on which the foregoing derivation is based requires that the condition that is to be avoided, namely, gear tooth separation, is assumed to occur and then the subsequent tendency of the acting forces is examined as to whether it results in increasing or decreasing the assumed gear tooth separation. While this assumed separation appears to be a mere artifice of analysis, it may actually occur even in gearing in which the foregoing inertia relations are satisfied, as a result of the building up and breaking down of an oil film between the teeth in question. Thus, when the inertia criterion, Equation 24, is satisfied, it insures that whenever such minute gear separation as that of the thickness of an oil film does occur, it will not increase to the full backlash of the gears. Steady equilibrium of forces is thus maintained at constant low worm velocities and a smooth chatterless descent of the load is realized.

The Correctness of these results has been verified on the

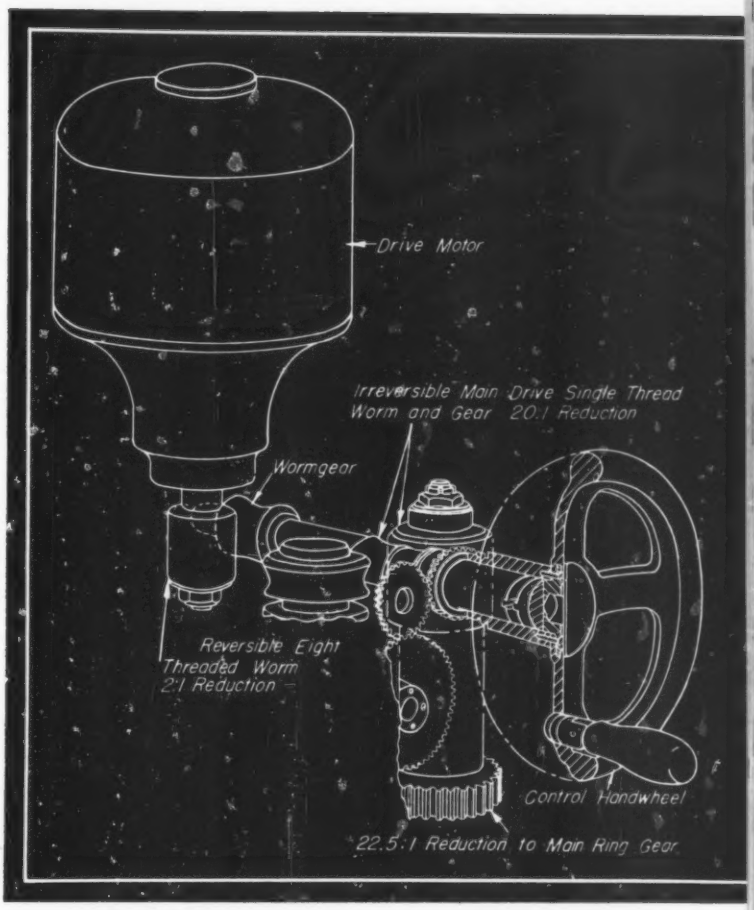

Drive Motor
Irreversible Main Drive Single Thread Worm and Gear 20:1 Reduction
Wormgear
Reversible Eight Threaded Worm 2:1 Reduction
Control Handwheel
22.5:1 Reduction to Main Ring Gear

Fig. 5—Stair-stepping on gun turret drive ceased after flywheel was added to shaft of irreversible worm. Addition of equivalent inertia to shaft of reversible worm was much less effective in preventing jerky operation

previously mentioned gun-turret drive whose stair-stepping originally led to this investigation. The turret in question weighed 12,000 pounds and had a radius of gyration of 36 inches. The overall speed ratio from worm to turret was 450 to 1. I_w originally was negligibly small compared to I_p/N^2, μ was about .12 at creeping speeds, and the worm lead angle λ was 4⅜ degrees = .081-radian. At low and creeping speeds μ was greater than λ and the criterion of Equation 25 was not satisfied, resulting in pronounced stair-stepping. The drive was made satisfactory by the addition of a small flywheel to the worm shaft, sufficiently large to cause criterion 25 to be satisfied. By Equation 25, the worm inertia requirements of this example, for 20-degree pressure angle gearing, are

$$I_w \geq \frac{12000 \times 36^2}{450^2} \times \frac{.12 - .081 \times .968}{.081 \times .968}$$

$$= 40 \text{ pounds-inches}^2$$

Thus, in this case a 5-inch diameter flywheel weighing 13 pounds was sufficient to assure smooth following of the 12,000-pound turret.

At higher speeds an oil film was established between worm and gear and μ became less than λ. In that condition, the drive functioned smoothly even with negligible

worm inertia, as required by the theory presented herein.

One important caution to observe in applying these results is that the inertia I_w must be rigidly connected to the irreversible worm shaft so that its acceleration forces are transmitted to the point of irreversibility with as little delay as possible. It was found that an equivalent inertia coupled to the worm shaft through other (reversible) gearing is not as effective in curing stair-stepping because of the inevitable end play, backlash, and shafting flexibility of the connected gearing which delay and weaken the transmitted inertia forces. Thus in the gun-turret drive cited, which is shown in *Fig.* 5, the gearing consisted of a 2:1 step-down with an 8-threaded reversible worm, a 20:1 step-down with the irreversible pair, and a final 22½ to 1 with a spur gear pair. Addition of the stabilizing inertia to the shaft of the 8-threaded worm was first tried, but was much less effective than an equivalent inertia mounted rigidly on the shaft of the irreversible worm and as close to the point of irreversibility as possible.

Windage Too Can Cause Stair-Stepping

Other instances of stair-stepping worm gearing have come to the author's attention. In a certain aircraft turret drive that was perfectly balanced as regards gravity forces, stair-stepping was found to occur due to windage forces on the protruding guns whenever the drive was urging the turret in the direction in which the wind forces on the guns were attempting to drive it.

A more prosaic and rather homely example of stair-stepping was provided by a commercial photographer's camera stand. The camera was mounted on a swivelled platform whose tilt was controlled by an irreversible worm gear drive with the motive power supplied by a manually operated crank on the worm shaft. The worm gearing was admittedly crude and exposed and without lubricant. Nevertheless, the camera could be smoothly cranked upward against the gravity force, but showed pronounced stair-stepping when it was driven in the opposite direction. The addition of a flywheel to the worm shaft, figured in accordance with criterion 25, would undoubtedly have made this drive equally satisfactory for both directions of camera adjustment.

Other Means for Eliminating Trouble

In closing, it may be well to mention that there exist other means for elimination of stair-stepping. One is to mate the gearing without backlash. Stair-stepping leads to gear tooth separation and the whole foregoing analysis was based on that fact. Obviously, if gear teeth can be maintained in contact, by whatever means, stair-stepping will not occur. However, this expedient is more of theoretical than of practical interest, since stair-stepping has occurred in gearing with tooth clearance held to .001-inch. To avoid stair-stepping by accurate meshing of gears would require the contacting surfaces to be positioned to an accuracy comparable to the thickness of a lubricating oil film, or about .0001-inch. And even if such accuracy were realized at the start, normal gear wear would soon spoil it.

Another, and perhaps the most obvious, device for elimination of stair-stepping is to load the output inertia with a biasing spring that is under a setup sufficient to balance the useful load force P. When the net force acting on I_w (or on M in the case of the inclined plane) is made sufficiently small, the tractive force F required to initiate motion of I_w (or of M) becomes correspondingly small and the starting transient accelerations of the system are correspondingly reduced. Where the load unbalance P is large, however, as in the case of gravity-unbalanced antiaircraft gun drives, the balancing spring must then be so large as to increase unduly the weight and space requirements of the mechanism. Moreover, where P is variable, as with gun elevation, the spring balance requires a compensating linkage that further complicates things and makes this expedient decidedly less practical than the addition of a small flywheel to the worm shaft.

Possibilities of the Heat Pump

A YEAR-ROUND air conditioning system known as the "heat pump" employs refrigeration equipment in such a manner as to enable the condenser to deliver usable heat to a space during one cycle of operation and to deliver cooling during the reverse cycle. The heat pump is not new, nor is it revolutionary. Interest in its possibilities, however, is increasing. It consists of a compressor, condenser, cooling coils, expansion valves, reversing valves and connecting piping—the same equipment, with the possible exception of the reversing values, that is normally used in a conventional mechanical refrigeration system.

When operating as a heating system, the cooling coils or evaporator absorb heat from an outside medium and deliver it, together with the heat equivalent of the work of compression, to the condenser which in turn gives it up to the space to be heated. On the other hand, when operating as a cooling system, the cooling coils or evaporator absorb heat from the conditioned space and reject it together with the heat equivalent of the work of compression, to the outside.

A number of such installations have been made; in the United States there are now in service perhaps 30 to 40. But the fact that standard items of heat transfer and refrigeration equipment can be employed is perhaps one of the principal reasons why no substantial progress has been made in the application of the idea to residential heating and cooling. The explanation of this lies in the fact that standard items put together simply could not be made to give either the performance or the low cost required for broad market development. Why the development work needed to remedy this has not been forthcoming is a question beyond the scope of the present discussion. However, there are indications that this condition is not likely to prevail much longer. One manufacturer, for example, is at present building units for home application which he expects to place in service immediately on completion. Although the quantity is not large, the prospect appears excellent that others will follow—and in much greater number.

From a paper by Philip Sporn and E. R. Ambrose, American Gas & Electric Service Corp., presented at the recent annual meeting of the A. S. M. E.

Applying Electric Heaters to Machines

By John W. Greve

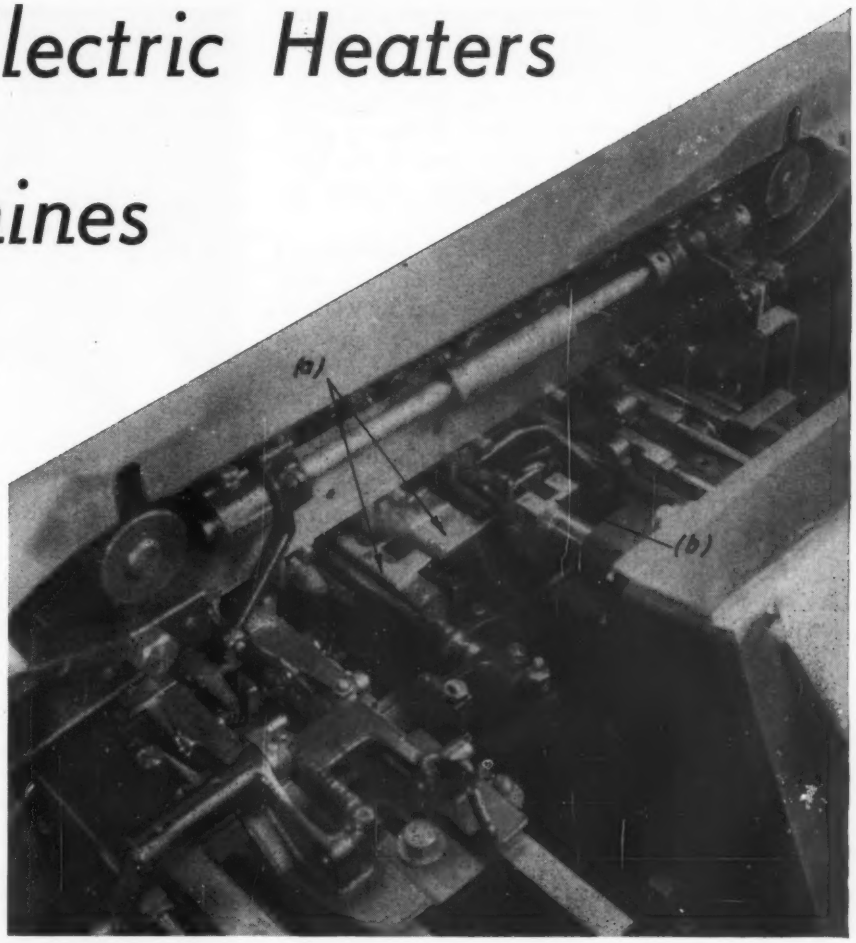


Fig. 1—Above—Precise control of heat is required to seal cellophane in cigar wrapping machine of the chain conveyor type. At (a) *are shown heated jaws of the sealer for the longitudinal seam and at* (b) *is the sealer for one end*

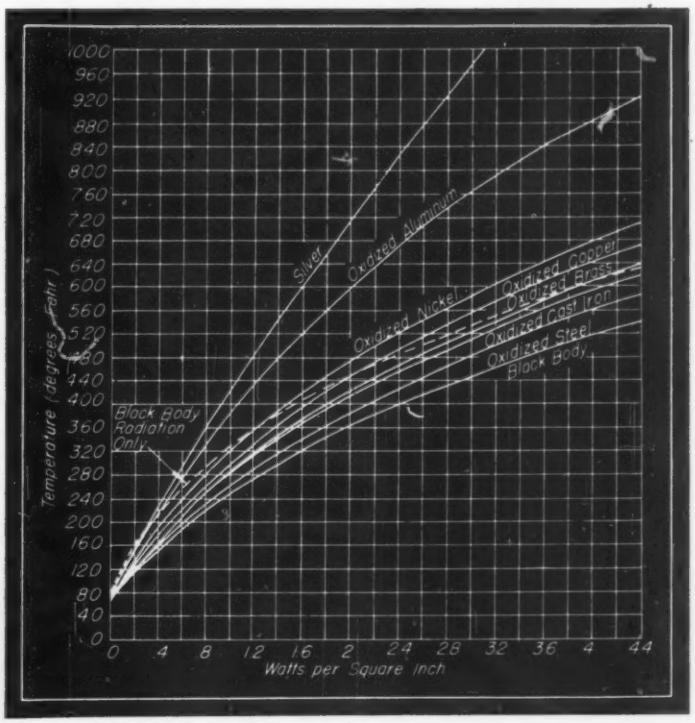
WHERE heating units may be needed, whether for processing, heat-sealing packages or ny of the myriad other applications, rip, cartridge or tubular electric eaters are finding increased favor. his results from their many advanges. Modern machines require pre-se, exacting control of heat which is adily obtained electrically. Also, requirements for maintenance and the resent need for self-contained production machines that are readily inalled or easily moved from location location are met better through the ilization of electric heating elements an by any other method.

Chief advantage of electrical heating elements is at the precise amount of heat can be produced exly where it is most desirable. This factor is also e of their major limitations unless the application designed to the requirements. In steam heating, instance, the piping or coils give up the heat of steam only at the locations where heat is being ithdrawn from the heated member whereas electric ating elements deliver the same quantity of heat unit length of heater unless, of course, a special ater is used. Therefore it is necessary to degn the application carefully to assure proper opation. If exact heat requirements are not known may be necessary to provide for varying the energy livered to the units or to conduct experimental or velopment work to ascertain the optimum comnation of heater capacity and location.

Oversize Heaters Are Usually Applied

In general, oversize heaters can be applied so that e heat required may be controlled through the use external resisters or multitap transformers. Where e fluctuations of heat might be too great for on-off ntrol with a thermostat, it is advisable to have the rmostat control only part of the heater load, the maining heaters being connected at all times as

Fig. 2—Below—Heat losses from vertical smooth surfaces of various materials. For horizontal surfaces 110 per cent of curve value may be used for top and 55 per cent for bottom



the basic load. Also, automatic regrouping of heater terminals to reduce input is very practical, such as delta-to-Y, or series-to-parallel, thus distributing the reduced heater load evenly. Heating elements should always be mounted so that the heat can flow as easily as possible into the work. This is done in the high-speed, intricate cellophane wrapping machine for cigars in *Fig.* 1.

In selecting heater size the heater requirements are seldom known precisely. Radiation and other losses, rate at which heat is given up to work in process, etc., are difficult to calculate. An approach to the problem, however, may be made by dividing it into two parts: The amount of Btu's required to bring the workpiece up to temperature and the amount needed to maintain the temperature with the heat being transferred to the process. For both phases of the calculation the radiation and convection losses should be added. These may be approximated by reference to *Fig.* 2, showing losses for smooth surfaces of various materials when in a vertical plane. If the surface is in a horizontal plane the losses from the top of a surface is 110 per cent of the curve values and those from the bottom are 55 per cent. Average losses from top and bottom surfaces as well as pipe surfaces are 82½ per cent of curve values.

Calculating Heat Requirements

Heat requirements for bringing the workpiece body and materials in process up to temperature may be calculated from their specific heat values and volume being heated in a given time cycle. These values in Btu's may be converted to kilowatts by dividing by 3412. As mentioned previously more heater capacity is usually applied than the minimum requirement, about 30 per cent being usually considered sufficient. Where possible, it also is advisable to use more heater units than actually required, thus lowering their operating temperature and increasing their life. Life of heaters depends on their watt-density and the frequency of on-off cycle. As these are reduced, life is extended.

Terminal connections on strip heaters are available in various standard arrangements. Both connections may be on one end, staggered, normal or parallel to the heater length, one on each end, or both in the middle of the strip length. Some heaters have three terminals with the middle terminal dividing the resistance element in two for three-heat operation. Cartridge heaters usually have both terminals or pigtails at one end. They may, however, have a terminal at each end. Strip heaters may be obtained in lengths from 4 inches to 96 inches whereas cartridge units vary from 1½ inches to 40 inches depending on diameters employed which range from ⅜ to 3½ inches. Tubular type units are available in lengths up to 10 feet with range in diameters from 5/16 to ⅝-inch.

Ring units, a modification of strip heaters, are available in sizes from 12-inch diameter down to small disks. Other special forms of strip heaters in-

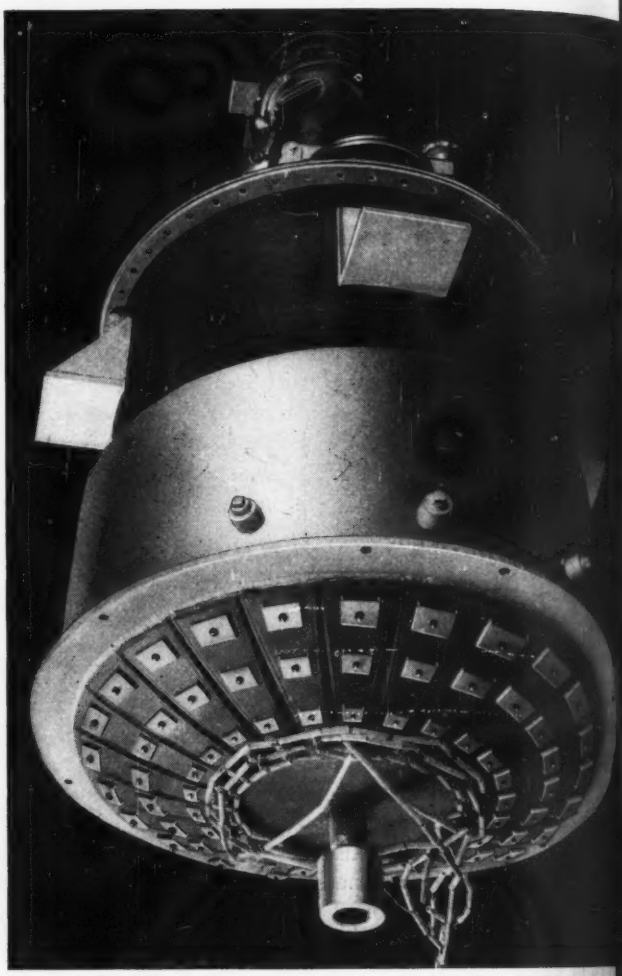
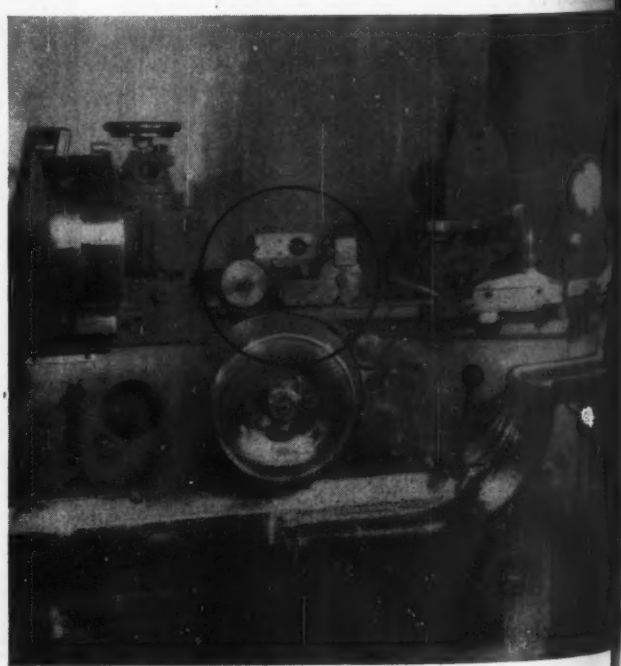


Fig. 3—Above—Segment heaters are formed to curvature autoclave bottom and fastened by welded studs to assure contact for heat transfer

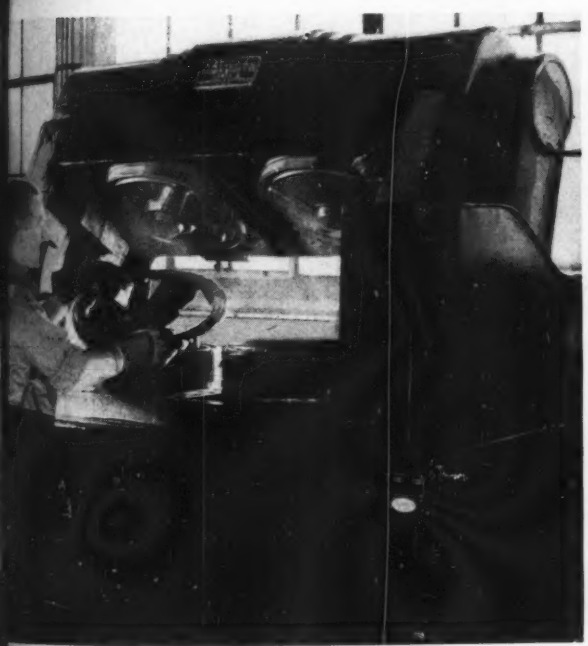
Fig. 4—Below—Heat sealer on cigarette-making mach utilizes cartridge unit mounted in a block between two pull A narrow steel belt running over the pulleys transfers the to the cigarette rod



de formed-segment strip heaters shown on the bottom the autoclave in *Fig.* 3. The strip heater elements are inches long, formed to the curvature of the tank to ure effective heat transfer. Each of the 24 segments rated at 1 kilowatt and controlled by automatic temrature regulation. Channel-iron plates fastened to ⅜-ch studs welded to the tank bottom clamp the heaters the vessel. An insulated sheet metal bottom cover proles an air space to increase heating efficiency.

Temperatures up to 1500 degrees Fahr. may be obned from tubular heaters. The standard strip heaters ving steel sheathing and nickel-chrome-iron resistors ould not be applied to temperature applications where sheath temperature would be above 750 degrees. For mperatures up to 1250, high-grade nickel-chrome alloy sistors are used with chrome-steel or stainless sheaths. onel sheaths are applied in humid atmospheres.

Standard tubular units have copper or brass casing and y be formed to shape for immersion heaters. These ings are suitable for heating water. For oil and alka-



g. 5—Plastic-molding press utilizes preformed tubular heaters cast into the aluminum alloy molds

e solutions steel casings are employed whereas Everdur, onel, or resin-coated casings are used for acids and other rosive agents. Seamless strip units are also used for mersion heaters.

Heat energy may be transmitted to the heated part in e of four ways. These are:

1. Conduction or contact
2. Radiation
3. Convection
4. Circulating system or forced convection

henever possible the first method, heating by conducis preferable because it is the most direct. Highest iciencies are possible and a minimum of materials, expting for radiant heating, are required. Also, design proions are simplified.

Choice between cartridge and strip heaters is usually determined by the application and the manufacturing facilities available. Cartridge heaters, however, are somewhat more costly. For heating by conduction both types are dependent upon good contact between the heater sheath and the heated body. Recesses for cartridge heaters are preferably drilled and reamed to a close assembly fit. Recommended clearance is .004-inch per inch of diameter. Expansion resulting from heating assures good contact throughout the heater length. The better the fit that will just allow assembly, the more efficient the heater application. The other extreme may be such poor contact as to result in low heat transfer with consequent overheating of elements and premature failure. Where possible it is best to provide a knockout hole by drilling through the body so that the heater may be removed easily should replacement become necessary.

Where suitable precision facilities are not available or the hole depth is too great for close fitting of the cartridge, an aluminum alloy or other material is poured between the cartridge and hole to form a bond for heat transfer. For this type of assembly it is necessary to select a matrix material that may be poured at a sufficiently low temperature as not to damage the heater. Also in case of maintenance replacement the matrix should be easily melted. On the other hand it should remain solid at the highest operating temperature of the unit. Because of their nature cartridge heaters are applied where the mass of the body requiring heat is sufficiently thick for drilling and where straight lengths of heater units may be utilized.

Sealer Utilizes Heat-Transfer Belt

An interesting application of a cartridge heater to a cigarette making machine is shown in *Fig.* 4. The sealer unit which dries the seam of the endless cigarette rod before it is cut into individual cigarettes consists of a narrow steel belt running over two pulleys. A cartridge mounted in a block between the pulleys heats the belt. The heat is controlled by a thermostat on top of the unit.

Cartridge heaters do not lend themselves to applications requiring special shapes but tubular heaters may be formed to any shape desired. For this reason they are utilized for immersion heaters or where heaters are formed and cast in platens, die blocks and other parts. *Fig.* 5 illustrates such an application. The press molds are cast aluminum alloy plates 19½ inches in diameter and 1¼ inches thick. Each top and bottom section is heated by a 4-kilowatt tubular unit preformed and cast into the molds to provide uniform heat distribution.

Flexible types of tubular heaters, employing lead sheathing, have been developed by a number of companies. Lengths are available in standard sizes of 60 or 120 feet for 110 and 220-volt service, respectively, delivering about 7 watts per foot with a maximum temperature rating of 165 degrees Fahr. Another type of "serpentine" unit, having coiled-wire sheathing, is tailored to the particular application. This type is shown applied to a cylinder in *Fig.* 6 and is suitable for temperatures up to 1000 degrees Fahr., consuming 15 to 20 watts per foot of length. The groove shown is turned in the cylindrical surface and the heating unit lies flush with the top of the grooving. A protective cover is shown partially covering the surface. Because of the flexible nature of the unit good contact is ob-

tained, fastening clips being used only at the ends.

Strip heaters may be applied to a split platen or block by milling slots in the split section, placing the heaters in position and assembling. Although an expensive operation, this is an excellent method from the viewpoint of neatness and efficiency, giving a clamping support to the heater throughout its length to assure good contact. Also there are no direct losses from the heater by radiation.

Strip heaters, however, are usually clamped to the back of a platen or heating shoe, often utilizing

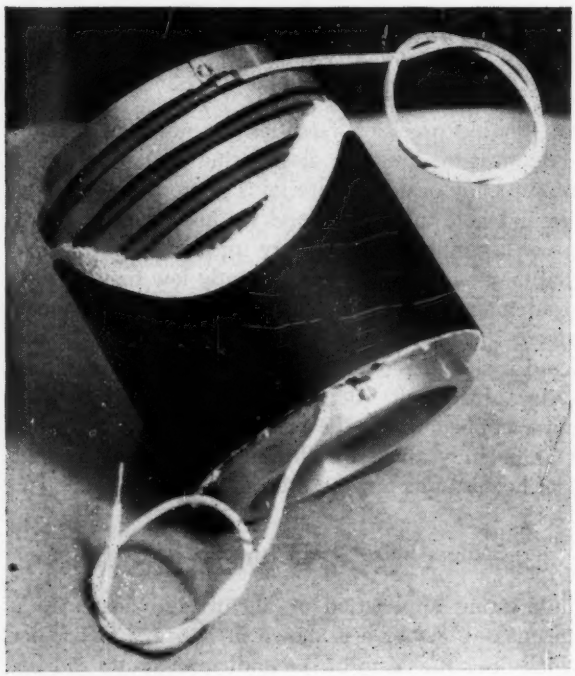


Fig. 6—Cylinder heater employs serpentine-type tubular units. Grooves allow the unit to be flush with the surface and clips are needed at each end only

milled slots to assure better contact. The clamps are placed relatively close together throughout the heater length to maintain good contact and to minimize any tendency to warp or bow away from the heated part. In this connection, if strip heaters are placed longitudinally around a cylinder or drum, clamps are preferable to bands or hoops whenever they can be applied because the heating cycle would tend to loosen the bands and reduce the area of contact even though the bands might have sufficient tension when the unit is assembled.

For drums like those mentioned, special heaters may be procured in which the contact surface of the sheath is concave to fit the curvature of the drum. If possible, however, standard flat-type strip heaters should be applied, though these require milling of flats on the drum. The choice of method should be dependent on costs, allowing a factor for possible need for replacement in the field and the facility with which standard heaters may be obtained compared with special shapes. If the drum is not large, strip heaters may also be advantageously applied in sections to form hoops, *Fig.* 7. These also are special shapes and their selection should be the result of analytical study. In all strip heater applications an allowance of ⅛-inch per foot of length should be made for expansion.

Another application indicating the versatility of electric heaters to precise heat control where space is not available for bulky equipment is the bread wrapping machine in *Fig.* 8. Adjustment provisions for the heated side plates also dictate the use of compact units. On each side the folder and sealing plates are heated by one 150-watt, two 400-watt and two 300-watt strip heaters. The bottom plates, shown in the foreground, are similarly heated to form the bottom seal on the wrapper. For loaves with concave bottoms a row of heated sealer fingers is provided followed by a row of unheated fingers to furnish additional pressure to the seal. Sufficient heat to melt the wax on the wrapper

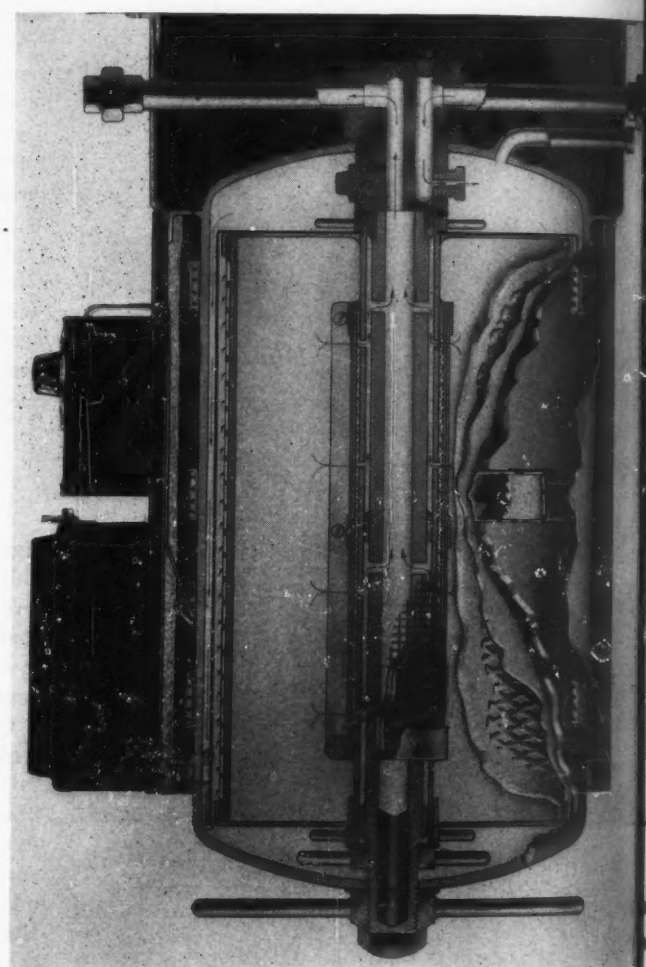


Fig. 7—Strip heaters formed in the shape of hoops around filter maintain working temperature of 120 degrees Fahr.

is applied by these units to make the seal as the loaves pass through the machine.

Radiant heat applications of these types of heaters do not have wide utilization. Where radiant heat is required, open coil heaters rather than the sheath-type heaters are satisfactory. They are less expensive and have the advantage of large areas of heater surface for a given space. Where only low temperatures are required, however, strip or tubular heaters have been used to advantage. They are easy to apply, may

(*Concluded on Page* 176)

Fig. 1—Overhanging load of aircraft generator imposes severe stresses on bracket and bracket fastenings

By A. M. Wahl

Westinghouse Research Laboratories

Effect of Vibration on Brackets, Fastenings

Fatigue strength of screw fastenings for bracket-mounted overhanging loads has been found to be a limiting factor in design. This is because the flexibility of the bracket rim imposes flexural stress on the fastenings, resulting in early failure. Tests on this phase of design were reported by the author at the recent annual meeting of the American Society of Mechanical Engineers, from which report this article is abstracted

BECAUSE of the severe vibration present in modern aircraft powerplants, engine-mounted auxiliaries must be designed to withstand accelerations of the order of 40 to 50g without failure, while at the same time the weight must be kept to a minimum. When attached units are overhung like the generator in *Fig.* 1, these vibrations impose severe reversed bending moments and fatigue loading on the bracket for attaching the generator to the engine.

To check the strength of brackets under such conditions, fatigue tests have been made on different designs and materials including forged steel, aluminum alloys, cast-steel alloy, and cast-iron alloy. This discussion will be

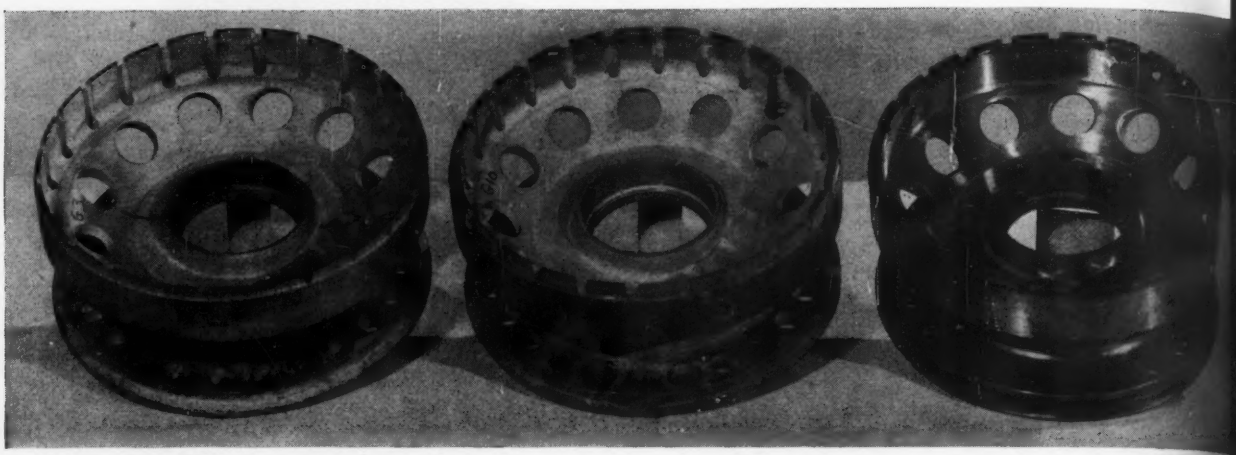


Fig. 2—Above—Typical forged-steel brackets tested by fatigue loading to study bracket design and its effect on bolt strength

limited mainly to the fatigue tests on the forged-steel brackets. These tests disclosed important design points.

When the series of fatigue tests was first begun it was soon found that in many cases the limiting factor in the design was not the fatigue strength of the bracket itself but rather that of the screw fastenings used to hold the bracket to the generator frame. Although it is frequently assumed that, if a screw or bolt is tightened to a load greater than the applied external load, fatigue failure will not occur; this is true only if the connected members have a sufficiently high rigidity. It is not the case where considerable flexibility is present. As Almen[1] has shown, the variable stress in such screw or bolt fastenings increases with the flexibility of the connected parts and, if this flexibility is too great, the variable stress present may be high enough to cause eventual fatigue failure regardless of the initial bolt load. In the case of the generator brackets tested, the flexibility of the bracket rim allows the screw heads to bend, thus setting up a variable stress which, when magnified by the stress concentration present, may result in fatigue failure. The strength of the screwed joint is thus dependent upon the rigidity of the bracket rim and will vary for brackets with different rim thicknesses.

The forged-steel brackets were made either from normalized SAE 1045 or heat-treated SAE 4140. Typical brackets of this type are shown in *Fig.* 2. Because of the thin rims used to save weight, the screw

Fig. 3—Below—Failure of screws for various bracket-rim thicknesses plotted against applied moment. In practically all cases the first screws to break were those farthest from the neutral surface of the bracket

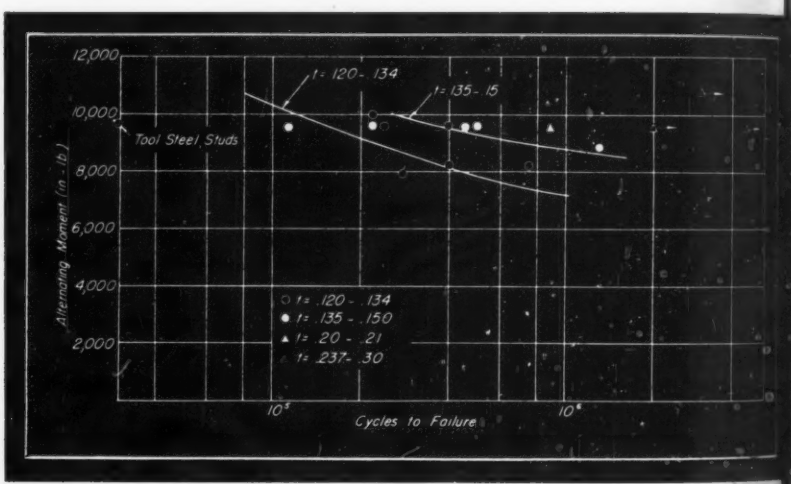
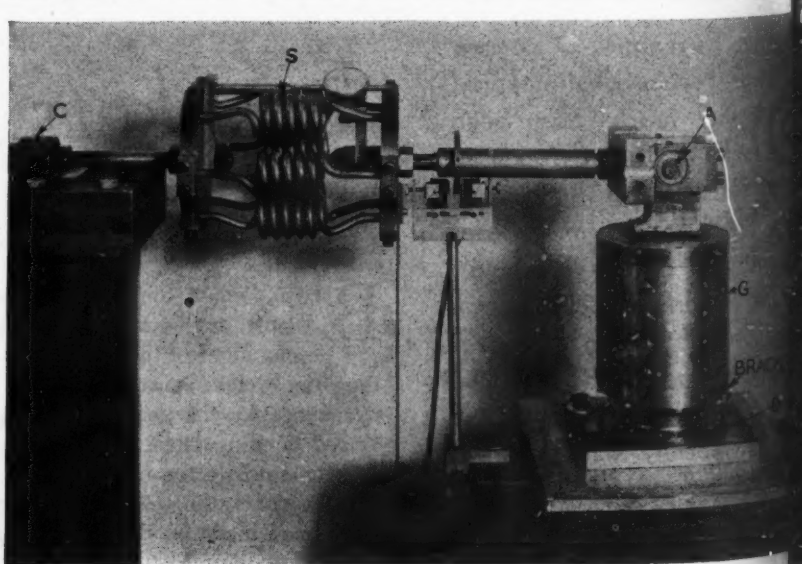


Fig. 4—Below—Method for fatigue-testing generator brackets, stimulating loading conditions. Alternating load is applied through calibrated tension-compressing springs. Loading is measured by dial and indicator on spring



"Tightening Is Vital Factor in Bolt Endurance"—J. O. Almen, MACHINE DESIGN, Feb., 1944. "Fatigue Failures in Common Machine Parts"—J. O. Almen, *Metal Progress*, Vol. 43, 1943, Pages 737-740. "Bolt Failure as Affected by Tightening", MACHINE DESIGN, Aug., 1943.

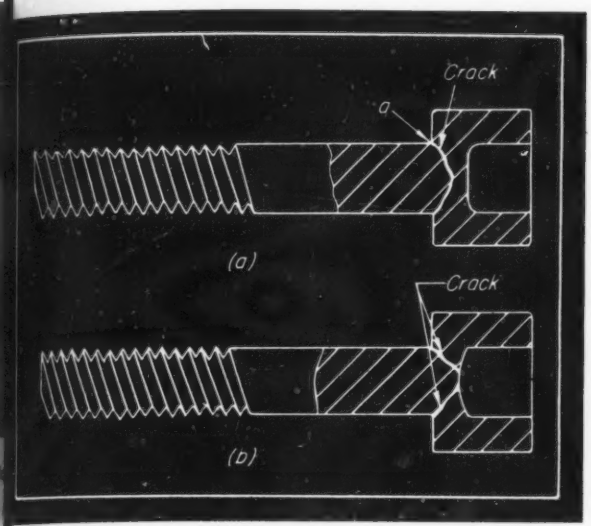


Fig. 5—Above—Typical shape of fatigue fracture of socket-head screws

Fig. 6—Below—Special design of bracket with flexible rim

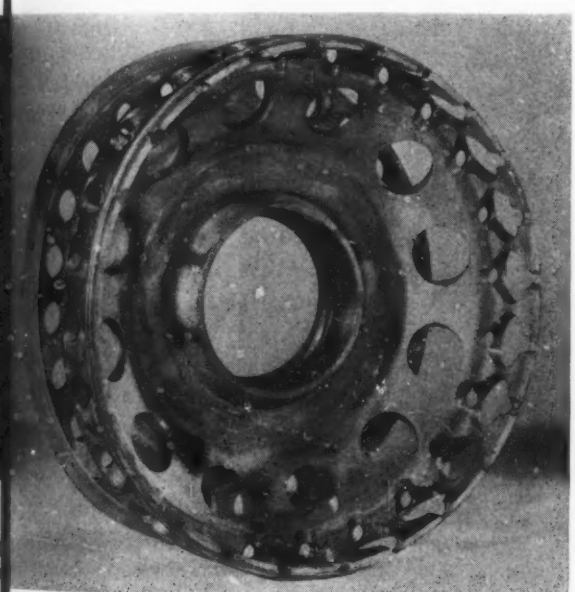
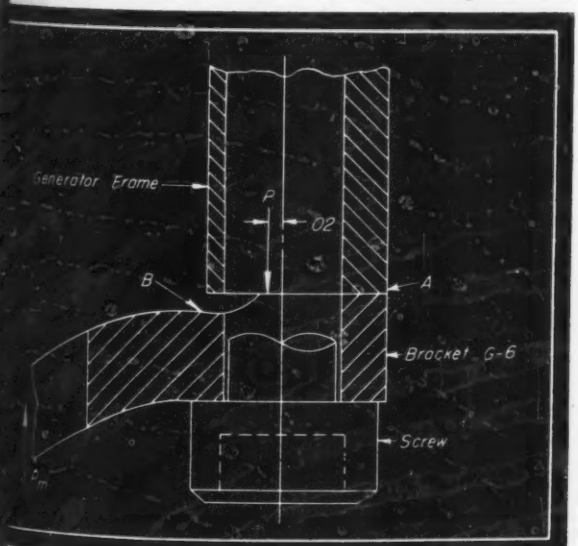


Fig. 7—Below—Load conditions for a flat-ring bracket



holes in most cases broke through the rim section. This, however, was not the case for one bracket—shown on the right in *Fig.* 2—which had a full rim, the hole diameter being less than the rim thickness. A flat-plate type of bracket will be discussed later.

In the fatigue-test apparatus for these tests, *Fig.* 4, the generator frame with attached bracket is mounted on a plate to simulate the attachment of the generator to the engine. An alternating load (reversed in direction) is applied through tension-compression springs, driven by a variable-throw eccentric mounted on a motor shaft. By this means an alternating bending moment may be applied to the bracket to simulate moments developed in service under severe vibration conditions. The load is determined by measuring deflections of the calibrated springs, using a dial indicator. For carrying out tests at elevated temperature (250 Fahr.), an electric heater is placed around the specimen.

In most of the tests 10-32 socket head screws (.19-inch diameter) were used to fasten the bracket to the generator frame and these were tightened to a wrench torque of 65 inch-pounds. Assuming a coefficient of friction of .25, this gave a calculated initial load approximately one-half the ultimate breaking load of 3200 pounds. The calculated working loads in most instances were only a small fraction of this value.

Bracket Assembly Dependent on Screw Strength

As indicated previously, when the fatigue tests of the forged-steel generator brackets were first begun it was found that the strength of the bracket assembly was limited by the fatigue strength of the screws. Since this in turn is dependent largely upon the rigidity of the bracket rim, the strength of the screwed joint will vary for the various brackets tested, depending upon the rim thickness. In practically all cases, the first screws to break were those farthest from the neutral surface of the bracket where the bending action of the spokes is the greatest. In general, the breaks occurred by snapping off of the screw heads, *Fig.* 5. In some cases the fatigue crack extended from the fillet directly across the body of the screw, while in others the crack extended through to the socket in the head of the screw.

The results with regard to screw failures are shown graphically in *Fig.* 3, in which the alternating bending moment is plotted against cycles to failure for various brackets with different rim thicknesses. (Only screw breakages are considered here, failures of the spokes or rim being discussed later.) Rim thicknesses varying from .12 to .134-inch are represented by open circles in *Fig.* 3, thicknesses from .135 to .15-inch by the filled circles. Although there is considerable scatter, the trend of the curves may be represented by the full lines shown. These give an indication of the increase in average fatigue strength of the screwed joint as the rim thickness is increased. This is because the variable stress in the screws is reduced as the rim thickness (or rigidity) is increased. Thus, for example, for a rim thickness of .145-inch, the calculated screw stress is 64,500 pounds per square inch at 9600 inch-pound moment; this drops to about 11,000 pounds per square inch for a rim thickness twice as great.

A test was made also on a bracket with a special flex-

ible rim, *Fig.* 6, obtained by drilling ⅛-inch holes laterally in the rim as shown. Failure of one screw occurred in a very short time, the fatigue life of the bracket being less than one-tenth the life of a similar bracket without holes in the rim. This result is, of course, to be expected because of the high variable stress in the screws resulting from the extreme flexibility of the rim.

The effect of variable stresses in the screws is also shown by the test on the flat-ring type of bracket. This bracket showed poor results, failure occurring after only 20,000 cycles at 9600 inch-pound moment. This may be explained as follows: The loading conditions at the screw joint of this bracket are as shown in *Fig.* 7. When the shear load p_m on the spokes acts upward as indicated, a resultant compression load P acts as shown between bracket and end of generator frame. Because of the short lever arm (estimated as about .02-inch for this bracket) a high load is developed, calculations indicating that the load on the screws is only a small fraction of that required to avoid opening of the joint at the applied moment used in the tests. Consequently, a high variable stress in the screws, due to joint opening, is induced and this accounts for the poor showing of this bracket.

Some improvement in the performance of this bracket might have been obtained by thickening the section near point *B* (*Fig.* 7) so as to maintain contact all along the generator frame thickness, and by using a larger number of screws; but even then calculations indicated that the forces developed will approach the values equal to the initial screw loads. Hence there would still be danger of loosening of the joint with consequent high variable stress in the screws.

Tests also were made under many different conditions including the use of different screws—standard fillister-head screws, screws with lower hardness, cadmium-plated screws, tool-steel studs, screws with special fillets under head, etc. However, as previously indicated, it was found that in most cases the big effect in determining fatigue strength of the screws was not the screw material but rather the rim rigidity of the bracket.

The nominal variable stresses in the screws for the various tests at different applied moments were calculated. These stresses are plotted versus number of cycles to failure in *Fig.* 8. (In this figure, a few points obtained from other tests on cast carbon-molybdenum steel brackets have also been included.) As may be seen, there is considerable scatter in the results, but an endurance limit somewhere around 25,000 to 30,000 pounds per square inch nominal calculated stress in the screws (or an endurance range of 50,000 to 60,000 pounds per square inch) is indicated. The endurance range of the screw material (hardness around 400 to 425 Vickers) should be around 180,000 to 200,000 pounds per square inch in reversed stress; a somewhat lower value would be expected with high mean stress.

Since the screws were subject to a fairly high initial stress as a result of tightening, this would appear to indicate a fatigue-strength reduction factor somewhere around 3. Because of the small fillet under the head, such

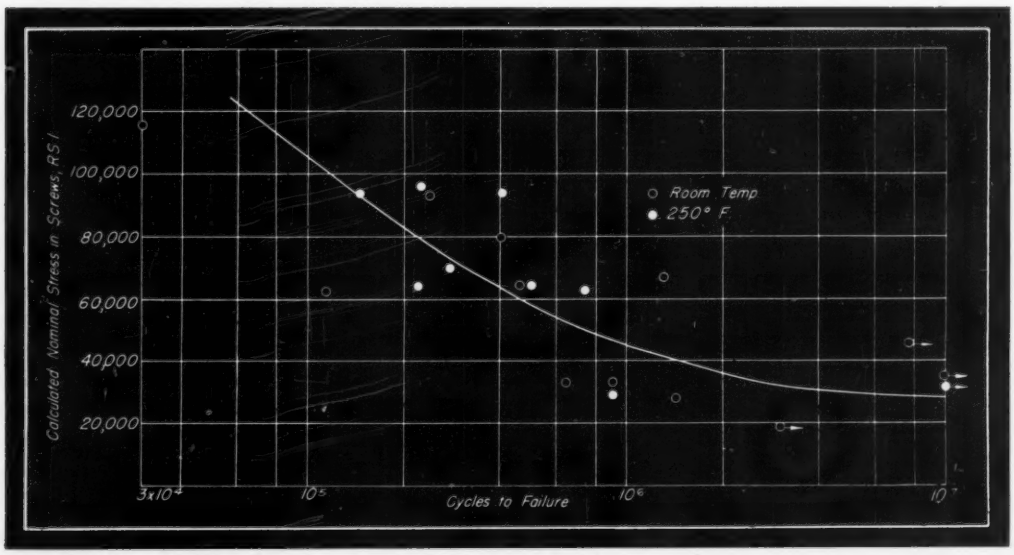
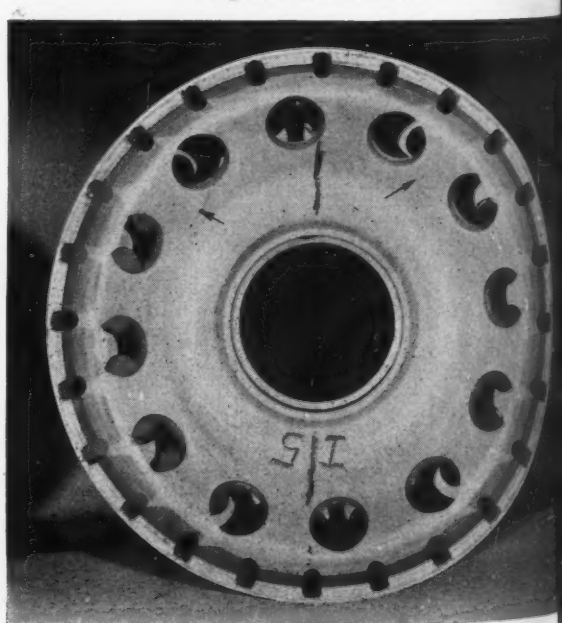
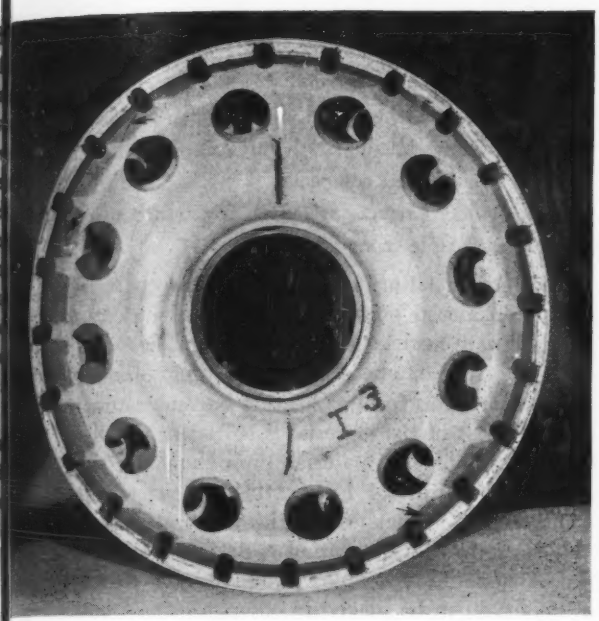


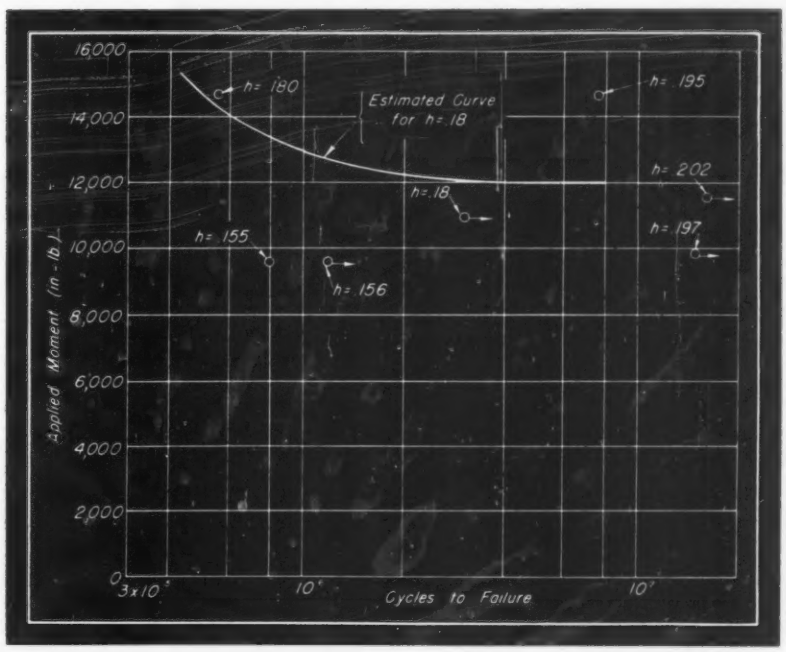
Fig. 8—Left—Calculated stress in screws versus number of cycles to failure for bracket fasteners

Fig. 9—Below—Fatigue cracks, indicated by arrows, in forged steel bracket





ig. 10—Above—Bracket having fatigue cracks in rim and spokes



ig. 11—Right—Spoke-failure curve for rged steel brackets. Screw failures are not included

tigue reduction factor appears reasonable, lthough on the basis of the stress-concen- raction factors obtained by photoelastic sts on T-heads by Hetenyi[2], considerably igher theoretical factors would be expected r small fillets.

The previous discussion has been con- rned mainly with screw failures since ese were a limiting factor in many es. However, bracket failures, usually in the spokes t sometimes at the screw holes in the rim, were o obtained in the course of the various tests. An illus- ation of fatigue cracks in a typical bracket is shown in g. 9, the direction of the applied moment being indi- ted by the black marks. It is of interest to note that in is case the fatigue cracks did not start at the spokes rthest from the neutral surface, but at a spoke farther wn. This indicates a higher stress at this point than at e top spoke where the bending action may be expected be greatest.

This was found by the brittle lacquer (Stresscoat) tests ried out by Hetenyi and Young on similar brackets[3].

[2] "Some Applications of Photoelasticity in Turbine Generator Design" M. Hetenyi, *Transactions, A.S.M.E.*, Vol. 61, 1939, Page A-155.

[3] "Application of the Brittle Lacquer Method in the Stress Analysis of achine Parts"—M. Hetenyi and W. E. Young, *Proceedings of the ociety for Experimental Stress Analysis*, Vol. 1, No. 2, 1944, Pages 8-119.

The first crack in the lacquer did not start at the top spoke but at one farther down. The explanation for this is that, although the bending action is a maximum at the top spoke, the effects of direct shear and torsion are greater at spokes near the neutral surface. However, these Stresscoat tests also showed that there was not a great deal of difference between the stress at the top spoke and that at which the first crack started.

Fatigue cracks in the rim as well as in the spokes are shown by the cracked bracket in *Fig.* 10. The rim failures in this case started near holes, as indicated by the arrows, the reason for this being the stress-concentration effect of the holes.

A summary of the fatigue-test results, as far as bracket failures are concerned, is given in *Fig.* 11, in which applied moment is plotted against number of cycles to failure, screw failures not being included. Since in most cases spoke failures only were observed, the minimum spoke thickness is indicated on each point, while arrows at various points indicate no failure. In all cases, the applied moment was obtained by multiplying the load by distance from resultant to midpoints of the spokes.

On the basis of the various tests, a rough estimate of the endurance curve for the brackets of heat-treated, forged SAE 4140 steel is shown by the full line in *Fig.* 11, for a minimum spoke thickness of .18-inch. This indicates an applied moment of about 12,000 inch-pounds at the endurance limit for this spoke thickness; lower values, of course, would be expected for the brackets with thinner spokes.

Taking the mean radius $r_m = 2.43$ inches, thickness $h =$ .18-inch, and width of spoke $b = .70$-inch at point of failure, *Fig.* 12, length a from mean radius r_m to point where crack starts equal to .29-inch as obtained by measurement, the calculated spoke stress comes out to be 63,000 pounds per square inch at the endurance limit, assuming 12,000 inch-pound applied moment.

The stress in the rim near the screw holes was also cal-

culated for the heat-treated, forged-steel brackets with a rim thickness of .205-inch. Taking the thickness at the point where fatigue crack starts as .205-inch, the calculated nominal stress at the endurance limit (12,000 inch-pound) becomes 37,000 pounds per square inch . If one assumes a stress-concentration factor of about 2, as seems reasonable due to the presence of the screw holes, this would mean a variable stress of about 74,000 pounds per square inch. This figure, while approximate, explains why failures also occurred near holes in the rim in some of the tests.

Since the heat-treated brackets had an average hardness of around 350 Vickers, a tensile strength of about 160,000

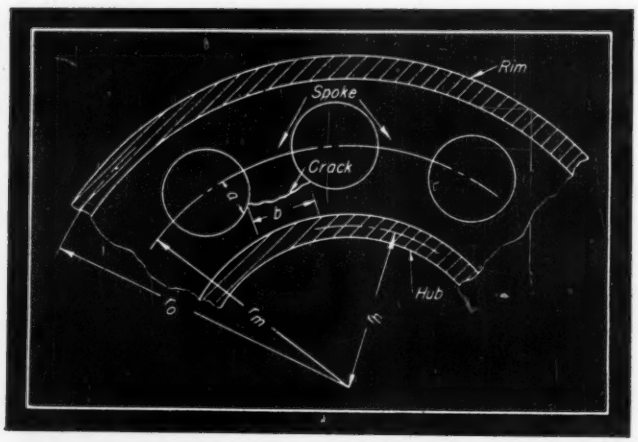


Fig. 12—Sketch showing fatigue crack in spoke and dimensions used for stress calculations

pounds per square inch would be expected. This would correspond to an endurance limit around 70,000 to 90,000 pounds per square inch[4]. The calculated figures for spoke and rim stresses previously mentioned, therefore, appear reasonable when the rough nature of the calculation is considered.

This investigation provides a demonstration of the fact that screw failures, in general, are due to variable stress, and that such failures may occur under fatigue loading provided the rigidity of the connected members is insufficient.

[4] "Manual of Endurance of Metals Under Repeated Stress"—H. F. Moore, *Engineering Foundation Publication* No. 13, 1927, Page 32.

Where Dielectric Heating May Be Applied

LIMITATIONS of dielectric heating which restrict universal application of this form of heating materials by a varying electric field are all interlocked to some degree with each other. These interrelations should be evaluated with respect to the actual application and have been summarized by Carl J. Madsen, Westinghouse electronics engineer.

High-frequency energy is not a cheap form of heat. It costs approximately two and one-half to four cents a kilowatt-hour, including direct operating cost, maintenance and amortization. Therefore it becomes economical only by virtue of one of the following factors:

1. Produces a result not possible by other methods
2. Increases the speed of the process, saving costs
3. Confines heating in the area needed
4. Is clean and compact
5. Reduces rejects or improves production quality
6. Is flexible in application.

Dielectric heating is applicable only to materials normally considered poor conductors of electricity or insulators. When electrical resistivity drops below 1000 ohm-centimeters, it may be necessary to apply other techniques.

In general this form of heating becomes attractive when the thickness of the material is sufficient to make difficult the heating of the material throughout by conventional means. In terms of inches this will vary with thermal conductivity. Thickness of the material must be reasonably uniform if even heating is desired. If the work is of such shape—for example, wedge shape—that two pieces of material can be placed in complementary positions and the resultant thickness be uniform, the material will heat evenly. Special electrode design and arrangement may sometimes be used when dealing with irregular-thickness materials.

Area to be heated limits the frequency which may be used in two ways: The capacitive reactance of a large area may become sufficiently low as to make tuning of the load extremely difficult if not impossible. The currents involved may also reach magnitudes difficult to handle. The length of the load may be such as to produce standing waves.

Since the amount of energy introduced into a material is proportional to frequency, the desire is to use as high a frequency as possible. The higher the frequency used, however, the greater the probability of unequal voltage distribution. If the longest dimension of the electrodes is a small fraction of a wave length, or if the material is scanned, this effect is minimized.

Another limitation occasionally encountered is that of generator design for high power at high frequencies. From a practical viewpoint it appears undesirable at present to use frequencies above 20 megacycles when power above 20 kilowatts is involved.

Because power input is proportional to the voltage squared, considerable power can be supplied at relatively low frequencies if the voltage is increased sufficiently. Two limiting factors apply. Total voltage between electrodes should be kept under 15,000. Precautions necessary to avoid corona and arc-over often are more costly than a compromise of another engineering factor. Voltage gradient permissible across the material may be a limiting factor. The radio frequency voltage which will puncture a given dieletric material is generally much lower than the 60-cycle voltage which a material will withstand.

As the loss factor, commonly used to express the degree to which materials will absorb energy, approaches zero it becomes increasingly difficult to transfer power into the material. Practical lower limit of the factor appears to be between .005 and .01. Upper limit probably blends into the range where the material becomes sufficiently conductive as to respond to inductive fields, although some skin effect will probably be noted before this limit is reached.

Analyzing the Strength of Curved Beams

By O. A. Carnahan
Professor of Mechanical Engineering
Syracuse University

Part II—Application of Equations

BASIC equations for the tangential, radial, shearing and combined stresses in curved beams were derived in the previous article, published in last month's issue. How these equations may be applied to the design of a curved beam member is explained in detail in the present article, from which it will be evident that the procedure is only slightly more laborious than for a straight beam member.

The following problem will illustrate the use of the formulas in designing a cast iron punch press frame.

PROBLEM: Design a box-section punch-press frame, as shown in *Fig.* 6, to withstand a load of 100,000 pounds with a throat depth of 36 inches, using working stress of approximately 10,000 pounds per square inch. If the designer has had no experience with frames of this type it may be necessary to make one or more trial solutions before arriving at a satisfactory section. This problem, however, will illustrate how much of the labor can be saved after the preliminary calculation has been made.

CALCULATION FOR TRIAL SECTION *O-A*: The trial section is shown in *Fig.* 7. R_{cg} (see Nomenclature, Part I) is first found by taking the moments of the areas about the center of curvature and dividing this by the sum of these areas. Referring to *Fig.* 7, the total area is made up of four rectangular sections as follows: (24 x 4) with its center of gravity (8+4)/2 or 6 inches from the center of curvature; two sections 1 inch x 24 inches deep with the center of curvature (8+32)/2 or 20 inches from the center of curvature; one section 20 inches x 2 inches with its center of curvature (34+32)/2 or 33 inches from the center of curvature. Location of the center of gravity for the whole section is therefore

$$R_{cg}=\frac{(24\times4)6+(2\times24)20+(20\times2)33}{24\times4+2\times24+20\times2}$$

$$=15.522 \text{ inches}$$

Radius to the neutral surface, R_o, is found by substituting in Equation 3. For these rectangular sections, $dA=bdr$, and the integral must be broken down to give

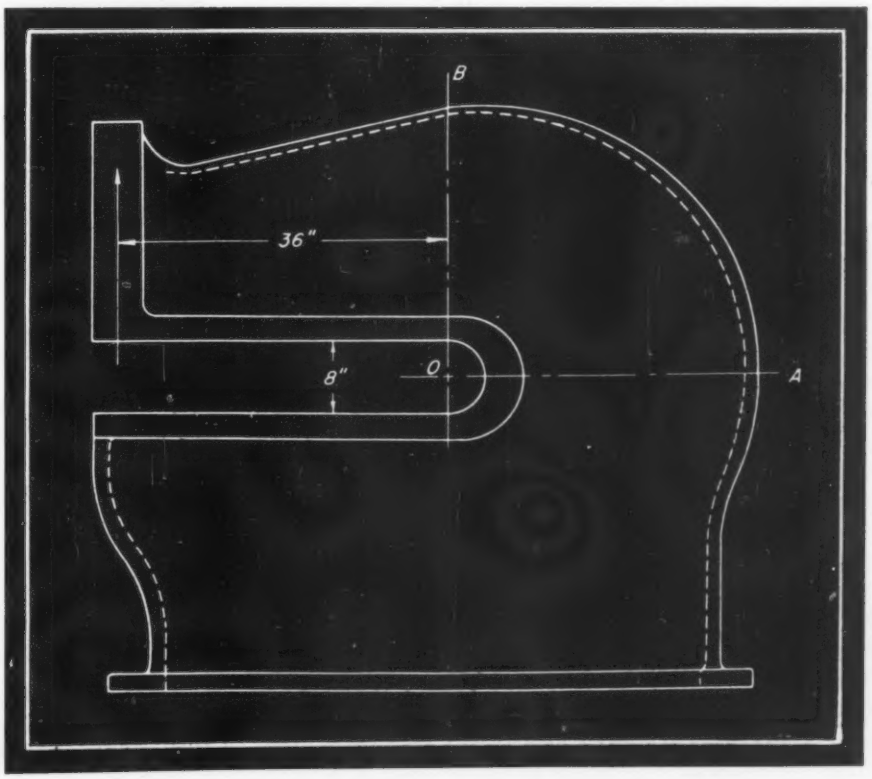


Fig. 6—Punch-press frame is a familiar application of a curved beam and is used in this article to illustrate the use of the formulas

$$R_o=\frac{A}{\int_4^8 \frac{24\,dr}{r}+\int_8^{32} \frac{2\,dr}{r}+\int_{32}^{34} \frac{20\,dr}{r}}$$

$$=\frac{A}{24\ log_e\frac{8}{4}+2\,log_e\frac{32}{8}+20\ log_e\frac{34}{32}}$$

Values of these logarithms may be taken from a table of Napierian (hyperbolic) logarithms, a log-log slide rule, or read from the graph, *Fig.* 3, Part I. It is also worth noting that $log_e(R_2/R_1)=2.3\ log_{10}(R_2/R_1)$. Substituting the values of these logarithms gives

$$R_o=\frac{184}{24\times.6931+2\times1.3863+20\times.0606}$$

$=8.924$ inches

Tensile stress at R_1, using Equation 5, is

$$S_t=\frac{100{,}000}{184}\left(1+\frac{(15.522+36)(8.924-4)}{(15.522-8.924)4}\right)$$

$=5768$ pounds per square inch

This stress is much below the allowable so the area may be reduced. By reducing all values of b by the same percentage, all values of r will remain the same and the stress will increase inversely with the area or breadth of the section. If the working stress is to be 10,000 pounds per square inch, the breadths may be made 5768/10,000=.5768 of the trial values. Using 9/16 (.5625) of the values of the b's, the corrected section is shown in *Fig.* 8. Then the new area =9/16 x 184=103.5 square inches, and the stress is $S_t=(16/9)$ x 5768=10254 pounds per square inch.

Radial stress at $R=8$ inches is found by substituting in Equation 7. It is to be noted again that $dA=b\,dr$, where $b=13\frac{1}{2}$ inches, while the final term in the numerator is evaluated only from $R_1=4$ inches to $R=8$ inches, or

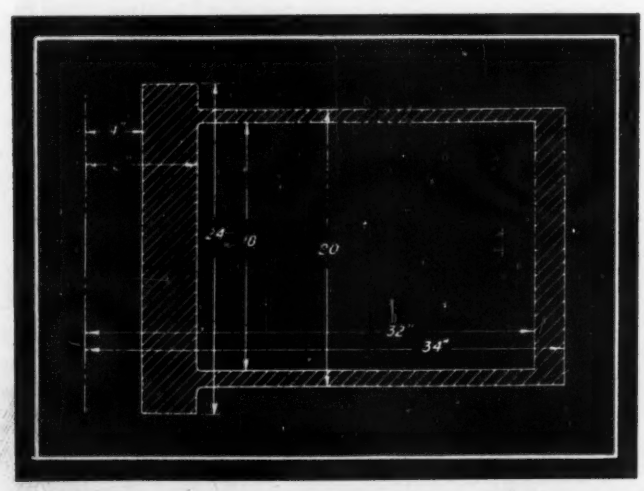


Fig. 7—Trial section in the horizontal plane OA is established on the basis of previous experience

$$\left[R_o\int\frac{dA}{r}-A\right]_{R_1}^{R}=8.924\times13\tfrac{1}{2}log_e\frac{8}{4}-13\tfrac{1}{2}\times(8-4)$$

$=29.5$

In the denominator of Equation 7, A is the total area of the section and B is the minimum width at $R=8$ inches (*Fig.* 8) which is 1⅛. Substituting in Equation 7,

$$S_{radial}=\frac{100{,}000(36+15.522)29.5}{103.5(15.522-8.924)\times8\times1.125}$$

$=24730$ pounds per square inch

This radial stress may be decreased by increasing the value of b at 8 inches from 1⅛ inches to 1⅛ inches x 24,730/10,000=2.782 inches. Rounding out to 2⅞ inches would make the radial stress at 8 inches equal to 27,820/2.875=9677 pounds per square inch. The fillet should be extended out from 8 inches to 8 + 24,730/10,000=19.784 inches. Making this 20 inches gives the final section shown in *Fig.* 9.

CHECK OF FINAL SECTION *O-A*: The new value of R_{cg}, from *Fig.* 9, is found as before, keeping in mind that the center of gravity of the triangular fillet is one

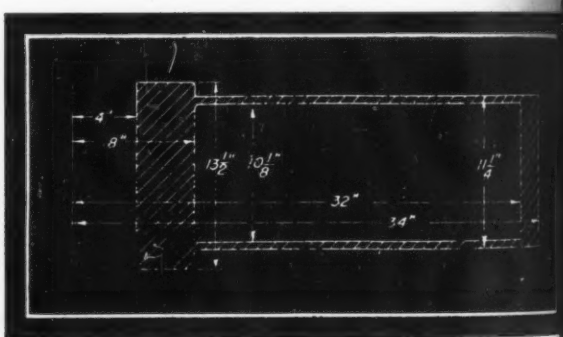


Fig. 8—Second trial section for horizontal plane OA modified from Fig. 7 to give correct tangential stress

third of the fillet length (or 4 inches) from the base of the fillet. Moment of the fillet area therefore is ½ 1¾ x 12 x 12=126 and

$$R_{cg}=\frac{(13\frac{1}{2}\times4)6+126+(24\times1.125)20+(11\frac{1}{4}\times2)33}{(13\frac{1}{2}\times4)+(1\frac{3}{4}\times6)+(24\times1.125)+(11\frac{1}{4}\times2)}$$

$=15.197$ inches

In calculating the new value of R_o from Equation 7 the term in the denominator corresponding to the fillet is

$$\int_8^{20}\frac{b\,dr}{r}=\int_8^{20}\frac{1\frac{3}{4}(20-r)dr}{12r}$$

$$=\frac{1\frac{3}{4}\times20}{12}\,log_e\frac{20}{8}-\frac{1\frac{3}{4}}{12}(20-8)$$

$=2.673-1.75=.923$

then, substituting along with values from *Fig.* 9,

$$R_o=\frac{114}{13\frac{1}{2}\log_e\frac{8}{4}+.923+1.125\log_e\frac{32}{8}+11\frac{1}{4}\log_e\frac{34}{32}}$$

$=9.104$ inches

Tangential stress at 4-inch radius is, from Equation 5,

$$S_t=\frac{100{,}000}{114}\left(1+\frac{51.197}{6.093}\times\frac{5.104}{4}\right)$$

$=10281$ pounds per square inch

Radial stress at 8-inch radius is, from Equation 7,

$$S_{radial}=\frac{100{,}000}{114}\times\frac{51.197}{6.093}\times\frac{9.104\times9.375-54}{8\times2.875}$$

$=10047$ pounds per square inch

Radial stress at 20 inches is found by integrating be-

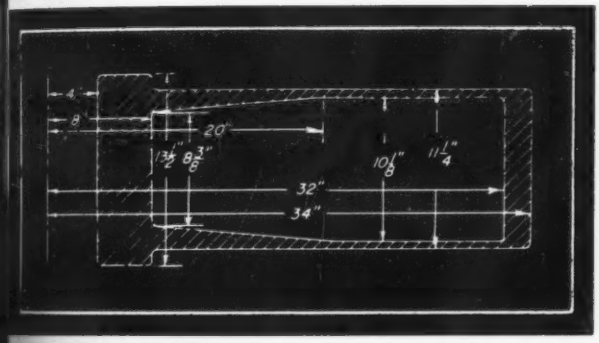


Fig. 9—Final section OA, modified from Fig. 8, takes proper account of radial stresses

tween 34 inches and 20 inches. The last term in the umerator of Equation 7 becomes, for this case,

$$\left[R_o\int\frac{dA}{r}-A\right]_{34}^{20}$$

$$=-9.104\left(11\frac{1}{4}\log_e\frac{34}{32}+1.125\log_e\frac{32}{20}\right)-$$

$$[11\frac{1}{4}(32-34)+1.125(20-32)]$$

$$=-9.104(.682+.529)-(-22\frac{1}{2}-13\frac{1}{2})$$

$=24.975$

hen, at 20 inches,

$$S_{radial}=\frac{100{,}000\times51.197\times24.975}{114\times6.093\times20\times1.125}$$

$=8180$ pounds per square inch

Calculations for Section *O-B*: Since the moment rm is 36 inches instead of 51.197, the moment will be bout .7 of the moment for section *A*, so the section may e reduced about 25 per cent in area if the total depth s about the same as section *O-A*.

Trial Section *O-B*, *Fig.* 10. Calculating R_{og} and R_o s before, the following results are obtained:

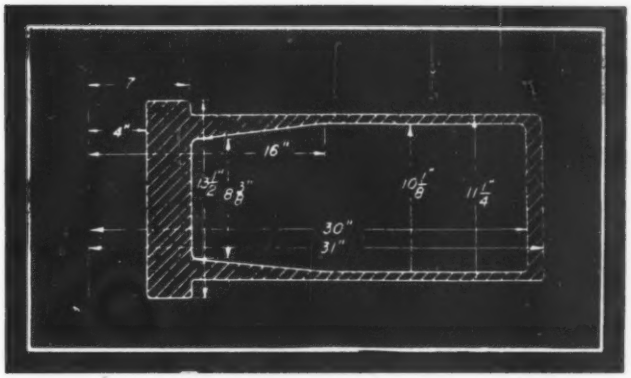


Fig. 10—Trial section for vertical plane OB, with slight modification, yields satisfactory stress values

$$R_{og}=\frac{222.75+78.75+478.69+343.12}{40.5+7.875+25.875+11.25}=13.137\text{ inches}$$

$$R_o=\frac{40.5+7.875+25.875+11.25}{7.555+2.572-1.75+1.637+.369}=8.235\text{ inches}$$

Substituting in Equation 8,

$$S_t=\frac{100{,}000}{85.5}\times\frac{36}{4.902}\times\frac{4.235}{4}$$

$=9100$ pounds per square inch

Combined stress at $R=7$ inches is found from Equation 13, in which the middle and last terms in the numerator are, respectively,

$$\frac{36}{2}+\sqrt{\left(\frac{36}{2}\right)^2+13.137^2}=30.3$$

$$8.235\times13.5\log_e\frac{7}{4}-40.5=21.7$$

and

$$S_t'=\frac{100{,}000\times30.3\times21.7}{85.5\times4.903\times7\times2\frac{3}{4}}$$

$=10800$ pounds per square inch

This stress S_t' can be reduced to the desired value by increasing the total value of b at 7 inches from 2⅞ inches to 2⅞ x 10,800/10,000=3.105 or 3⅛ inches, which corrects the 8⅜ inches dimension of *Fig.* 8 to 8⅛ inches. The fillet should also be extended out from 16 inches to (3.105/1⅛) x (10,800/10,000) x 7 inches=21 inches.

From the foregoing discussion the general procedure in designing a curved beam for minimum weight may readily be followed. It should be pointed out that the new formulas for obtaining radial and shearing stresses are of particular value in determining the strength of box, T or I-sections where the mass of material is concentrated near the inner and outer radii in order to make the frame as light as possible. Other sections such as circular, elliptical and trapezoidal are heavier, while the radial stresses in them never attain values approaching the strength of the material and therefore need not be considered.

Precision Casting Obviates Machining

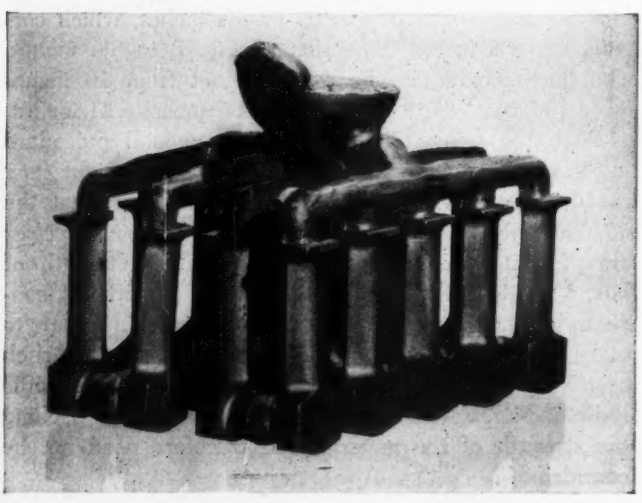
By Arnold Ristow
Allis-Chalmers Mfg. Co.

IN THE manufacture of aircraft turbosupercharger buckets, difficult metallurgical and machining problems arise due to the nature of the material that must be used to meet extreme conditions. Operating speeds of 25,000 revolutions per minute create enormous centrifugal forces which the buckets must withstand, while the exhaust gases driving the turbine are hot enough to raise the working temperature to 1600 degrees Fahr.

To add to the complexity of the problem, the gases do not reach the turbine blades in a continuous flow but rather in a series of closely spaced blasts, inducing tremendous vibrations and pulsations. Also, the gases are chemically corrosive and the high flow speeds cause a mechanical erosive action.

Few metals or alloys possess the combination of characteristics required, strength at high temperature being the most difficult to fulfill. After experiments with alloys using cobalt, chromium and molybdenum, an alloy of this class known as Vitallium has been found to be highly satisfactory. This alloy at room temperatures shows an ultimate tensile strength of 100,000 pounds per square inch with an elongation of 8 to 13 per cent. Yet at 1500 degrees Fahr. the tensile strength is still 65,000 pounds per square inch with elongation 20 per cent.

Although Vitallium satisfies the requirements, it is extremely hard and can be machined only by use of abrasive wheels and belts. Consequently "lost wax", or investment, casting which allows the buckets to be cast to within a thousandth of an inch of the finished dimension, proved to be an invaluable aid in their production.



In the process, bucket patterns are molded in groups of four, with individual riser and a common branch riser molded integral. Molding is done in bismuth-lead die sets made from a steel "master" machined to the exact shape of the bucket plus shrinkage allowance.

After clamping in the fixture, wax at a temperature of 140 degrees Fahr. is injected into the mold under a pressure of 65-85 pounds per square inch. Then the molds are opened and the wax pattern removed. Six of the groups are assembled with runners, risers and sprue to form a complete pattern for 24 buckets. The pattern assembly is dipped in a silica flour suspension, and extremely fine sand is sprayed over all the surfaces to back up the dip coat, after which the pattern is allowed to dry thoroughly for 48 hours.

Making the Mold

A chromium-nickel steel flask is employed. A liner of waterproof paper is put in place and the entire assembly waxed together, forming a mount that holds the pattern securely in place during further handling. Liquid mold material, called an "investment" and made from selected sands with a liquid binder, is poured around the pattern. The flask, filled slowly with a small scoop, is placed upon a vibration table to settle the fines in the mix and to pack them closely around all portions of the pattern.

After drying for eight hours, the molds are placed on a steel grating where steam is directed against them. The heating first melts the wax holding the base plate, allowing the plate to be removed, then melts out the wax.

Molds next are placed on alloy steel trays which are fed into a two-zone gas-fired pusher-type furnace, the first zone operating at 500-600 degrees Fahr., the second at 1850 degrees Fahr. The purpose of this heating is to vaporize any remaining wax in the mold and to raise the temperature of the mold high enough to assure fluidity of the Vitallium metal, preventing excessive chilling as the metal flows through the mold. Small electric carbon arc furnaces are used to melt stock for the castings.

To assure proper filling of the mold and aid in production of sound castings, air pressure is applied when the metal, at a slightly superheated temperature, is poured in the mold. After cooling for 4½ hours, the casting is removed from the mold. The investment is extremely hard, and the flask is first placed in a fixture positioned so that an air hammer can strike directly against the crucible head of the sprue. This hammer breaks up the mold to such an extent that the casting can be removed.

From this point the castings go to an abrasive cutting machine where the gates and riser are trimmed from the buckets. Investment still clinging to the buckets is knocked off as the buckets are tumbled and shot blasted

GASKETS IN DESIGN

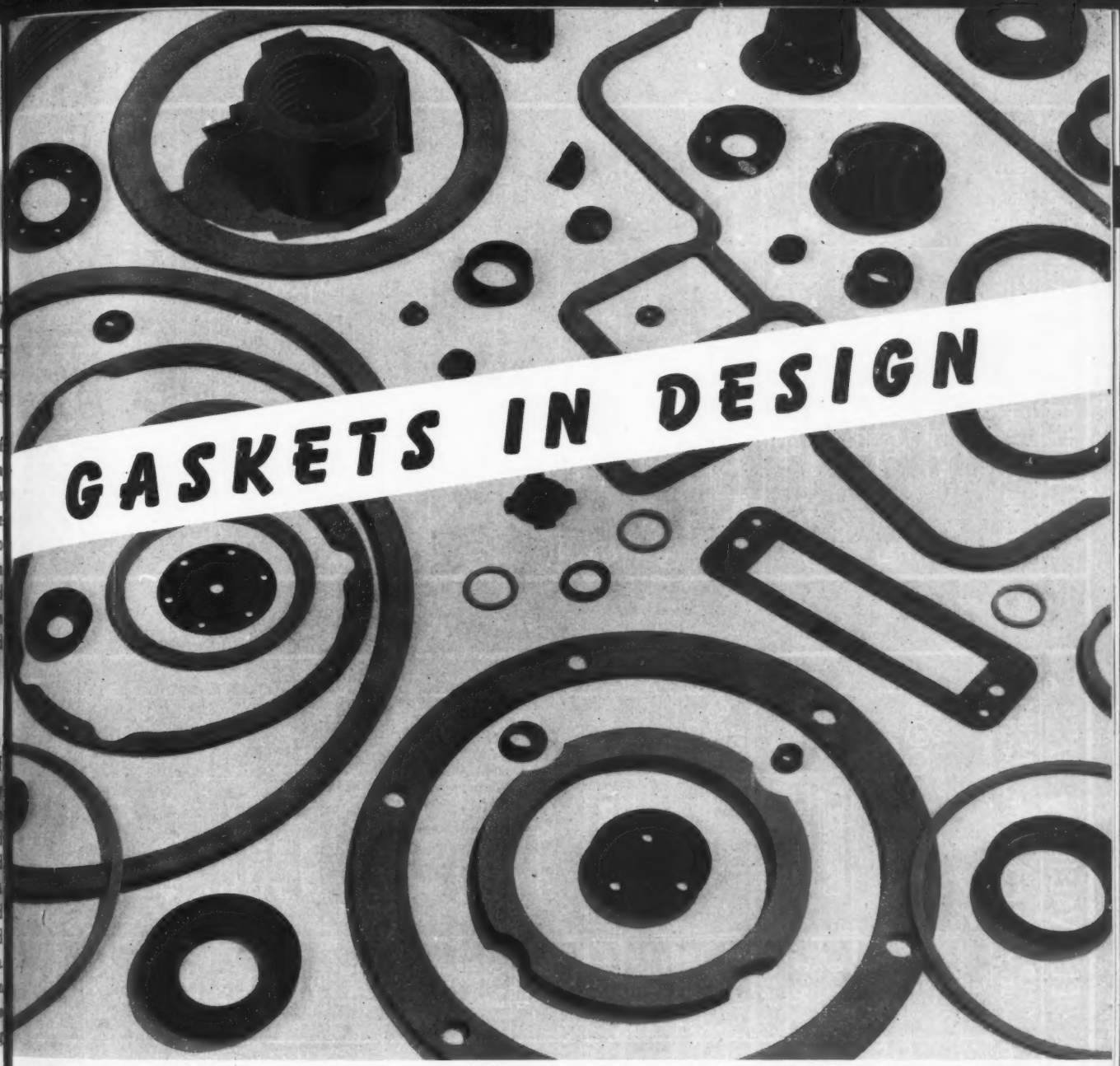


Fig. 1—Wide variation in gasketing materials necessitates careful design consideration

Roger W. Bolz

PROPER understanding of the basic factors involved in gasket selection can go a long way toward avoiding costly failures in gasketed installations. A practical method of approach that might be used in choosing a satisfactory gasket design from the great variety of types commercially available would, without doubt, eliminate the major portion of the problem. However, owing to the immense number of variables associated with the gasketing process which have not been completely evaluated, an absolutely reliable method of selection has not as yet been devised. In most instances, nevertheless, applications based on experience in the field will result in satisfactory service.

Of the numerous variations in gasketed joints and methods of gasketing, the compression type gasket is probably the most common. This article will be confined, therefore, for practical purposes, to the general steps that may be followed as an aid in the selection of compression gaskets, regardless of the nature of the equipment in which

Fig. 2—Below—Yielding of a gasket under pressure. Typical joint surfaces, the upper has serrations and the lower a plain finish

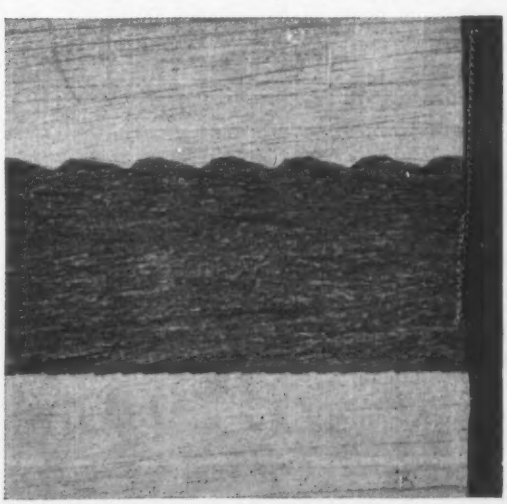


TABLE A: GASKET FACTORS m AND YIELD VALUES y

GASKET MATERIAL		GASKET FACTOR m	YIELD VALUE y	SKETCH & NOTES	REFER TO TABLE B: FACING LIMITATION	USE COL.
GUM RUBBER SHEET		.50	0		USE ①④⑥ ONLY	II
HARD RUBBER SHEET		1.00	180			
CLOTH INSERTED SOFT RUBBER		.75	50			
CLOTH INSERTED HARD RUBBER		1.25	400			
VEGETABLE FIBRE SHEET (HEMP OR JUTE)		1.75	1120			
RUBBERIZED WOVEN WIRE INSERTED ASBESTOS	3-PLY	2.25	2200			
	2-PLY	2.50	2880			
	1-PLY	2.75	3650			
ASBESTOS COMPOSITION OR COMPRESSED ASBESTOS	1/8" THICK	2.00	1620		USE ①④⑥ ONLY	
	1/16" THICK	2.75	3650			
	1/32" THICK	3.50	6480			
SPIRAL-WOUND METAL, ASBESTOS FILLED	CARBON STEEL	2.50	2880			
	KA2S OR TYPE 316	3.00	4500			
SERRATED STEEL, ASBESTOS FILLED		2.75	3650			
CORRUGATED METAL, ASBESTOS INSERTED OR CORRUGATED METAL JACKET, ASBESTOS FILLED	SOFT ALUMINUM	2.50	2880		USE ① ONLY	
	SOFT COPPER OR BRASS	2.75	3650			
	IRON OR SOFT STEEL	3.00	4500			
	MONEL OR 4-6% CHROME	3.25	5450			
	11-13% CHROME, KA2S OR TYPE 316	3.50	6480			
CORRUGATED METAL	SOFT ALUMINUM	2.75	3650			
	SOFT COPPER OR BRASS	3.00	4500			
	IRON OR SOFT STEEL	3.25	5450			
	MONEL OR 4-6% CHROME	3.50	6480			
	11-13% CHROME, KA2S OR TYPE 316	3.75	7600			
FLAT METAL JACKET, ASBESTOS FILLED	SOFT ALUMINUM	3.25	5450			
	SOFT COPPER OR BRASS	3.50	6480			
	IRON OR SOFT STEEL	3.75	7600			
	MONEL OR 4-6% CHROME	4.00	8820			
	11-13% CHROME, KA2S OR TYPE 316	4.25	10120			
GROOVED IRON OR SOFT STEEL CORE, METAL JACKETED	SOFT ALUMINUM	3.25	5450		USE ①④⑥ ONLY	
	SOFT COPPER OR BRASS	3.50	6480			
	IRON OR SOFT STEEL	3.75	7600			
	MONEL OR 4-6% CHROME	4.00	8820			
	11-13% CHROME, KA2S OR TYPE 316	4.25	10120			
SOLID METAL, 1/8" THICKNESS OR MORE (FOR THINNER GASKETS	SOFT ALUMINUM	4.00	8820	NOTE: FOR EACH 1/32" REDUCTION IN THICKNESS BELOW 1/8" INCREASE m BY 25% (COMPUTE		I
	SOFT COPPER OR BRASS	4.75	13100			
	IRON OR SOFT STEEL	5.50	18000			
	MONEL OR 4-6% CHROME	6.00	21780			

TABLE B: EFFECTIVE GASKET WIDTH b

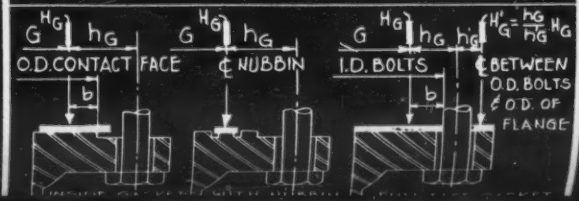
FACING SKETCH	BASIC GASKET YIELDING WIDTH b_o COL. I	BASIC GASKET YIELDING WIDTH b_o COL. II
1a	$\frac{n}{2}$	$\frac{n}{2}$
1b	$\frac{w+t}{2}\left(\frac{w+n}{4}\ \max\right)$	$\frac{w+t}{2}\left(\frac{w+n}{4}\ \max\right)$
2	$\frac{w+n}{4}$	$\frac{w+3n}{8}$
3	$\frac{w}{2}$; $\left(\frac{n}{4}\ \min\right)$	$\frac{w+n}{4}$; $\left(\frac{3n}{8}\ \min\right)$
4	$\frac{3n}{8}$	$\frac{7n}{16}$
5		
6	$\frac{n}{4}$	$\frac{3n}{8}$
7		
8	$\frac{w}{8}$	

EFFECTIVE GASKET YIELDING WIDTH b

$b = b_o$, when $b_o \leq 1/4"$

$b = \frac{\sqrt{b_o}}{2}$, when $b_o > 1/4"$

LOCATION OF GASKET LOAD ATTACK



hey are to be used or the details of the closure design. Selection of a gasket for a given installation is dependent upon the various features of the closure construction and the prevailing operating conditions. Mechanical features of the closure restrict the choice to those types of gaskets possessing suitable physical characteristics. Resistance of the gasket material to the effects of operating conditions further restricts this choice. All effort, therefore, must be directed toward that combination of materials and construction which will produce and maintain as near a perfect seal as possible under the specific operating conditions.

Specifications for a satisfactory gasket material might be narrowed down to a few major factors. In effect the gasket shall:

1. Withstand the necessary bolt pressure without crushing or excessive flow
2. Have sufficient resilience, without excessive thickness, to distribute the applied pressure evenly
3. Have sufficient elasticity to maintain an adequate amount of the applied pressure in operation
4. Resist sliding out in operation if not retained by a groove or shoulder
5. Be reasonably unaffected by the medium handled
6. Chemically resist on both inner and outer exposed edges, the mediums contacted
7. Not contaminate the medium or promote corrosion
8. Be easily installed or separated and removed.

Because the governing factors of closure construction and operating conditions are closely related, the engineer in selecting a gasket should first consider the expected operating conditions to determine the necessary flange construction. Contacting surfaces of the closure flanges can be finished to suit any gasket which may be chosen. Selection of the gasket should, as a rule, be made at the time that the joint construction details are determined.

Sealing of a gasketed joint by compression between flanges is effected through yielding, or flow of the gasket material into the various imperfections of the flange contact faces, resulting in a smooth unbroken barrier having no pathways through which the confined media may escape. The extent to which a gasket material must flow in order to create an impervious seal will depend upon the type of contact face and its surface finish. Obviously, less flow of the gasket material is required to seal a reasonably smooth surface than a rough one. *Fig.* 2 shows how adequate bolting flows the gasket into flange imperfections. Joint surfaces, in this illustration, are representative of ordinary machine work—the upper flange having typical serrations and the lower a so-called smooth finish.

The surface that appears to be best for most gaskets consists of a smooth, flat and clean machine finish, such as might be made by a 90-degree V-nose tool at a feed of about .01-inch. Serrated faces consisting of concentric V-shaped grooves spaced approximately 1/32-inch are commonly used with soft gaskets. The so-called "gasket-surface" consisting of a spiral groove made with a 1/16-inch radius round-nose tool at a 1/32-inch feed, provides a most difficult facing to seal, particularly with a metal gasket, although it is satisfactory if rubber or sheet packing is used. The standard "phonograph" finish consisting of a V-shaped spiral groove at 1/32-inch feed should never be used with metallic gaskets. Serrations greater than 1/64-inch in depth are not usually recommended.

The amount of force that must be applied to the contact area of the gasket to compress or flow the material is referred to generally as the *yield* stress. Theoretically, this value is considered to be independent of the internal working pressure and is, therefore, the minimum stress that must be applied to the gasket if a good seal is to be had, even with low pressures. However, application of internal working pressure may introduce a condition which tends to separate the flanges and lower this effective stress on the gasket. Ability of a gasket to maintain a seal at reduced stress is a function of the resilience of the material. The ratio of this so-called resultant or residual stress to the internal pressure is, for all practical purposes, a constant throughout a wide range of pressures for a given gasket construction and composition. This ratio is called the *gasket-factor* or *contact-pressure ratio.*

Physical characteristics of a gasket material determine the required yielding stress and gasket-factor, hence the amount of force that must be applied to initiate and maintain a tight seal. Since the majority of joints are assembled by means of bolts, the amount of bolting required must be calculated. The minimum total bolt load in pounds may be determined from the greater of the values obtained from the following formulas†.

Operating or working conditions

$$W_m = H + H_p = .785G^2p + 2b(3.14Gmp) \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Atmospheric temperature conditions without internal pressure

$$W_m = H_y = 3.14bGyr \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

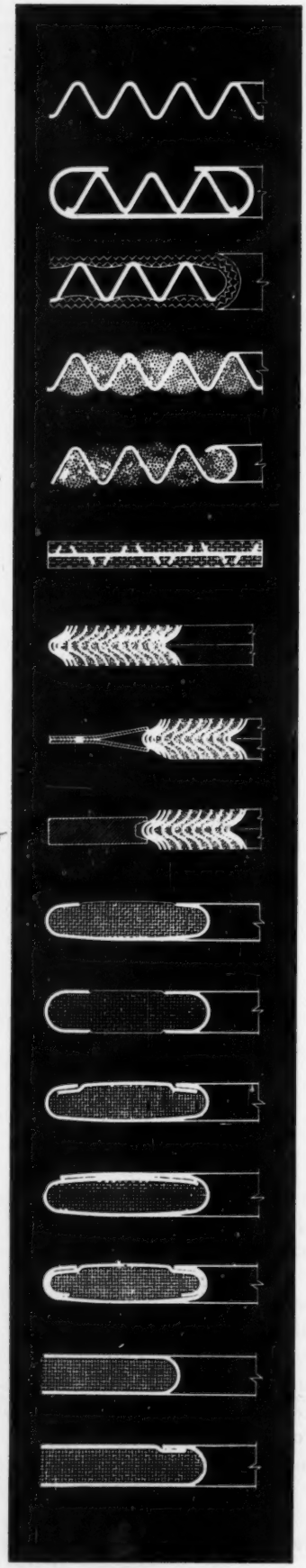
where

W_m = Minimum required bolt load in pounds
H = Hydrostatic end force in pounds
H_p = Total joint-contact surface compression load in pounds
H_y = Total joint-contact seating load in pounds
G = Mean diameter of gasket in inches
p = Maximum allowable working pressure in pounds per square inch
b = Effective gasket seating width in inches
m = Gasket-factor or contact-pressure ratio
y = Gasket unit seating load in pounds per square inch
r = S_b/S_a
S_b = Maximum allowable bolt stress at operating temperature
S_a = Maximum allowable bolt stress at atmospheric temperature.

Minimum bolt load determined by foregoing process will be sufficient to resist, under maximum operating or

†A.S.M.E. Unfired Pressure Vessel Code.

Fig. 3—Left—Gasket factors, yield values, basic yielding width and load-attack location for various designs and materials commonly used. (From "Gasket-Loading Constants" by Rossheim and Markl)



Drawing, courtesy Goetze Gasket & Packing Co. Inc.

TYPE 1. Plain type deeply corrugated metal

TYPE 2. Deeply corrugated metal core partially enclosed by a metal shell

TYPE 3. Corrugated metal core covered with woven asbestos cloth

TYPE 4. Corrugated metal with twisted and treated asbestos cord cemented into corrugations

TYPE 5. Corrugated metal with asbestos cord cemented into corrugations and with reinforcing inner lap

TYPE 6. Sheet steel core having clincher prongs embedded in asbestos sheets

TYPE 7. Interlocked plies of preformed metal strip spirally wound with cushion of asbestos strip

TYPE 8. Spirally wound type with centering guide for accurate positioning

TYPE 9. Spirally wound type with combination centering guide and compression limiting gauge

TYPE 10. Soft filler partially enclosed by a metal shell

TYPE 11. Soft gasket material reinforced by metal rims

TYPE 12. Soft gasket filler completely enclosed by a two-piece metal shell

TYPE 13. Soft gasket filler completely enclosed by a single metal shell

TYPE 14. Soft gasket filler encased by two reversed metal shells

TYPE 15. Soft gasket filler with single metal jacket

TYPE 16. Soft gasket filler with two-piece metal jacket

Fig. 4—Some of the most popular of the various types of gasket designs showing the cross-sectional construction

working conditions, the hydrostatic end force exerted by the internal pressure upon the area bounded by the mean diameter of the gasket. In addition, it will maintain a predetermined compression load upon the gasket which, experience has proved, is ample to assure a tight joint. The tables in *Fig.* 3 list many of the commonly used gasket materials and contact facings, with values of m, b and y that have proved satisfactory in actual service.

Actual bolt load W_a which must be at least equal to the minimum requirements found by equations 1 and 2, is the total force in pounds available when the total bolt area is stressed to the maximum allowable working stress at operating temperature. This is found as follows:

$$W_a = A_b S_b \quad \ldots\ldots (3)$$

where A_b is the total cross-sectional area of bolts at the root of the thread, or section of least diameter, in square inches.

For use in the design of the flanges or contact surfaces, the flange design bolt load W must not be less than the average of the minimum required bolt load W_m and the actual bolt load W_a or,

$$W = \frac{W_m + W_a}{2} \quad \ldots\ldots (4)$$

This provides a margin of safety against abuse or mishandling from such factors as overbolting. Should additional safety or application of the full available bolt load be desired, it is recommended that actual bolt load W_a be used as a basis for design.

In order to apply these mechanical design factors to best advantage, the influence of operating conditions upon the selection of gasket materials must be investigated. Here, primarily, the temperature and nature of the confined media limit the choice of materials. From the types available, illustrated in *Figs.* 4 and 5, three classifications might be made, depending upon ability of the material to resist heat. Nonmetallic gaskets, and those incorporating rubber or low melting-point metals, are generally limited to temperatures below 250 degrees Fahr. The semimetallic group composed of gaskets of asbestos, partially or completely jacketed with metal, may be used up to 850 degrees Fahr. For temperatures exceeding 850 degrees Fahr., only all-metal types should be used.

In general there are no pressure limitations on these metallic or semimetallic gaskets. Pressure failures with gaskets of metal are usually due to corrosion of the gasket, improper design, or maintenance of the closure. The temperature at which various gaskets may be used is restricted mainly by the materials and not the construction, except in designs subject to long-time relaxation or creep.

Last, but not least, the less decisive operating conditions which include the operating cycle, vibration, and the frequency with which the joint will be opened, must be examined in the light of their influence upon the efficiency of the gasket. Sudden variations in the operating conditions, or extreme variations in temperatures—inside or out—may require the use of an exceptionally resilient gasket material. The number of times a closure must be broken in service, may call for a gasket that can be reused many times without serious deterioration. Such se-

ndary effects of operating conditions cannot be disregarded, since they may have a decisive part in the overall maintenance of maximum joint efficiency. The final selection, of course, can then usually be made on a basis of relative costs, availability or other similar features.

Some of the mechanical considerations of the various types illustrated in *Figs.* 4 and 5 are:

TYPE 1: Although suitable for most operating conditions, this design is generally used for low-pressure installations where the flanges are lightly bolted. In some materials this design is subject to severe stress corrosion and creep at high temperatures.

TYPE 2: Generally made up with a corrugated core of relatively hard metal, jacketed with a soft metal. Useful for large tongue-and-groove joints that must be frequently opened. Core helps prevent creep of jacket at elevated temperatures.

TYPE 3: This construction has been useful in sealing low-pressure units where the flanges were particularly poor. Its use is limited by the composition of the impregnating compound with which the asbestos is treated.

TYPES 4 AND 5: Asbestos-insert corrugated metal gaskets are in general use for low-pressure gas and steam service. Particularly suitable for use with highly concentrated or hot acids when made up with blue asbestos cord. Type 5 is a modification of the standard for special service where asbestos must not be exposed to confined fluid.

TYPE 6: Reinforced asbestos millboard is suitable for sealing low-pressure gases in light flanges, such as furnace sections, air ducts, exhaust manifolds and cylinder heads.

TYPES 7, 8 AND 9: Suitable for all flange conditions, and because of its resilient construction, can readily be seated with relatively light bolting. This design is capable of withstanding severe shock conditions and will compensate for extreme expansion and contraction. It is not generally recommended for hot oil service.

TYPES 10 AND 11: So-called single-jacketed metal asbestos-filled gaskets are standard for handhole and most small diameter closures in auxiliary pressure equipment. This construction is used to seal spark plugs, valve bonnets, exhaust manifolds and the like.

TYPES 12 TO 17: These gaskets are perhaps the most popular of the metal-clad asbestos-filled types. The double-jacketed construction results in a totally enclosed gasket which is used for high-pressure, high-temperature applications in pipe flanges, pressure vessels and heat exchangers. The asbestos filler imparts a degree of resilience to the gasket, aids in compensating for inequalities in the flange faces and to some extent, for poor flange alignment. The modifications shown as 13 and 14 are for special installations. Type 13 is available only in very narrow widths where the regular Type 12 could not be made. The French type construction 15 is used for valve bonnets, cylinder heads and vacuum service, where smooth unbroken contact faces are required. This gasket construction, when made up with a thick soft filler, is finding extensive service in glass and glass-lined equipment in which the jacket material is likely to be made from silver or platinum. Types 16 and 17 are required when the gasket must be excessively wide.

TYPE 18: Inside-open metal-jacketed asbestos gaskets are used particularly to seal off threaded assemblies. The asbestos will not damage the threads. When used in a

TYPE 17. Soft gasket filler with three-piece metal jacket

TYPE 18. Soft gasket filler with single metal jacket over part of both faces

TYPE 19. Soft gasket filler completely enclosed by a corrugated metal shell

TYPE 20. Corrugated metal core reinforced by a corrugated metal shell

TYPE 21. Heavy solid metal with concentric grooves

TYPE 22. Heavy solid metal with spaced concentric ribs

TYPE 23. Heavy solid metal with twisted and treated asbestos cord cemented into concentric grooves

TYPE 24. Heavy solid grooved metal core with single metal jacket

TYPE 25. Heavy solid grooved metal core partially enclosed by single metal shell

TYPE 26. Heavy solid metal with spaced concentric ribs installed with compressed asbestos composition sheet

TYPE 27. Two metal plates with spaced concentric ribs welded together around outer periphery

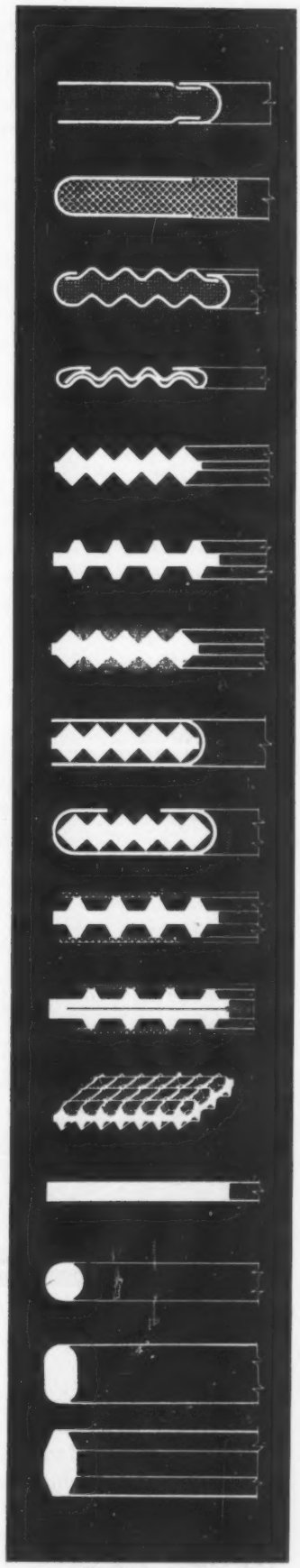
TYPE 28. Solid metal with raised cross ribs

TYPE 29. Plain solid metal or soft gasket material

TYPE 30. Solid round section metal or soft type gasket material

TYPE 31. Toroidal cross-section gasket of any solid metal

TYPE 32. Octagonal cross-section gasket of any solid metal



Drawing, courtesy Goetze Gasket & Packing Co. Inc.

Fig. 5—Additional gasket designs indicating the wide variety of constructions and combinations available

standard type closure, the internal pressure aids in maintaining a tight seal and jacket helps prevent blowouts. Not suitable for liquid service unless asbestos is treated.

TYPES 19 AND 20: Modifications of the Type 12 gasket, this construction has greater resilience and less contact surface. Long used in the oil industry, this design is finding applications in other fields. It is not subject to overstressing failures, and is less likely to creep at high temperatures than the plain corrugated metal design. Type 20 is a similar construction designed for service above 850 degrees Fahr., the upper limit for asbestos-filled gaskets.

TYPES 21 AND 22: Profiled and serrated gaskets are used where a solid metal construction of reduced contact surface is desirable. These designs can be more easily seated than a plain solid gasket and are used extensively in high-pressure steam lines.

TYPE 23: Additional sealing action of asbestos cord makes this type suitable for rough and uneven flanges.

TYPES 24 AND 25: Profile gaskets partially or totally jacketed with metal have the advantages of reduced contact area. For corrosive service an iron core jacketed with a resistant metal is commonly used in chemical and oil industries. Generally limited to widths of 1 inch.

TYPE 26: A modification of the serrated type, this gasket is more readily sealed and has somewhat better resilience. Used in high-pressure, high-temperature steam lines operating under adverse conditions.

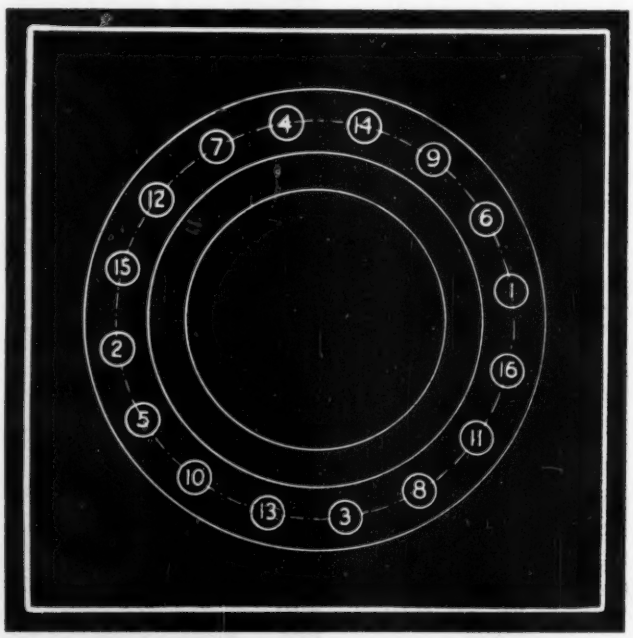


Fig. 6—Correct sequence of tightening procedure for bolted gasket contact surfaces

TYPE 27: Designed to expand under internal pressure, this gasket will maintain a tight joint under the most extreme operating conditions of pressure, shock and vibration. Automatically compensates for bolt relaxation.

TYPE 28: This construction has found application in certain hydraulic closures.

TYPE 29: The most common form of soft flat gaskets of any suitable material. Limited to low pressures and temperatures. Plain solid metal gaskets, although not as popular as they were, still are widely used for high-pressure service, particularly in tongue-and-groove joints. They generally require more unit stress to seat than any other gasket design.

TYPE 30: Round-section solid metal gaskets are used for special applications. The flanges generally are cut with V-grooves. Excellent for high-vacuum service.

TYPES 31 AND 32: Ring type joint gaskets of solid metal are almost universal throughout the oil industry for high-pressure, high-temperature service. The octagonal shape is slowly replacing the oval as it results in less damage to both the gasket and flange. Can be used only in flanges designed for them.

Proper provision for satisfactory installation cannot be over-emphasized. The gasket should be carefully centered. This usually is taken care of by a centering device or the gasket itself, but there is always some play determined by the bolt hole clearance, etc. This makes necessary accurate lining up of the edges of the contact surfaces and the gasket.

Proper sequence of bolt tightening is very important. A great deal of gasket leakage can be traced to improper tightening of the bolts. The diagram in *Fig.* 6 shows the correct sequence to be followed regardless of flange size. Amount of stress increase applied to the bolts as they are being tightened should be small at the start and raised only when the final desired stress is almost reached. Unless this precaution is observed, contact surfaces will be forced out of parallel. This may not be too serious with soft, thick gaskets, but when metal types are used, even slight out-of-parallelism can result in a leaking joint.

Judicious use of a torque wrench will help a great deal in obtaining uniformly stressed flange bolts. The mechanic can be given a series of torque readings and instructed to bring all the bolts up to each reading before going to the next higher torque. In this connection, the torque wrench should not be used to measure the exact bolt stress. The readings are, some believe, dependent upon too many variables for it to be considered a satisfactory stress indicator in these applications.

In the field of highly specialized applications, where metallic gaskets cannot be used due to lack of flange strength or chemical attack, synthetic rubbers are often found to be excellent. Ordinary synthetic rubbers are satisfactory where temperatures do not exceed 300 degrees Fahr. Synthetic rubber and asbestos compounds may be used where temperatures reach as high as 500 degrees Fahr. Recent developments in the field of hydrocarbon and vinyl plastics promise to furnish the designer with gasket materials ideally suited to almost any operating condition. Chemical resistance equivalent to that of many synthetic rubbers is combined with flexibility and toughness over an extremely wide temperature range in these new plastics. The new synthetic rubber-like silicones and another plastic—acrylic polymer—are proving valuable for gaskets in such items as super-chargers and searchlights where the extreme temperatures resulted in early failure with previously used materials.

MACHINE DESIGN acknowledges with appreciation the collaboration of the following companies in the preparation of this article: Armstrong Cork Co. (head illustration *Fig.* 1); Garlock Gasket & Packing Co. Inc. (*Figs.* 2, 4, 5 and 6); Garlock Packing Co.; Hycar Chemical Co.; M. W. Kellogg Co. (*Fig.* 3).

Simultaneous Hardening and Tempering

By Stephen Smith
Air Reduction Sales Co.

Fig. 1 — Above — Tank bogie wheel rim is rotated past hardening and tempering flames with water quenching jet in between

THOUGH flame hardening had made great strides prior to our entry into war its scope had nevertheless been limited with respect to the degree of surface hardness that could be obtained on ferrous metal. It has not been possible to produce at will, until recently, any desired degree of surface hardness with the basic methods of flame hardening.

Established metallurgical principles of heat-treating practice, such as heating the surface above the critical temperature, quenching with a suitable medium, and tempering or reheating to the proper temperature to produce the desired degree of surface hardness, are incorporated in a new process of simultaneous flame hardening and tempering. What the technique really does is to combine heating, quenching and tempering into a single, progressive operation.

At first it would seem that the development of this method of heat treatment would be obvious. The radically new idea in the three-in-one operation is the fact that the tempering process is conducted at the same speed as the flame-hardening operation and simultaneously with it. This performance is in direct contrast to the accepted and established practice of furnace tempering.

Fig. 2—Below—Production machine simultaneously hardens and tempers four large tank bogie wheel rims

Subsequent Drawing Obviated

Previously existing flame-hardening processes were limited to producing surface hardnesses close to the maximum obtainable on the material under treatment. Formerly, to obtain an intermediate or lower hardness, the fully flame-hardened surface had to be subjected to a subsequent furnace drawing operation. It is here that the new

technique deviates from the standard practice of flame hardening, though in principle they are analogous.

The material under treatment is progressively heated by a multiflame oxyacetylene tip, followed immediately by a quenching stream. This stream is followed at a predetermined distance by another softer oxyacetylene flame which heats the hardened surface to the proper drawing temperature. This technique permits any surface to be completely flame hardened in one continuous operation, to any desired degree of hardness.

Among the many problems with which the Army Ordnance department has had to deal is that of preventing the battering of rim flanges on tank bogie wheels. Army field tests had revealed, under simulated battle conditions, that the rim flanges became dented and twisted from contact with stones and other obstructions. This was due to the fact that the flanges did not have the protection of the hard rubber tires that are molded on the rims. The problem boiled down to this: How to give the necessary hardness to the exposed flanges and still retain the impact resistance desired.

In the accompanying illustrations the simultaneous flame hardening and tempering process is shown in op-

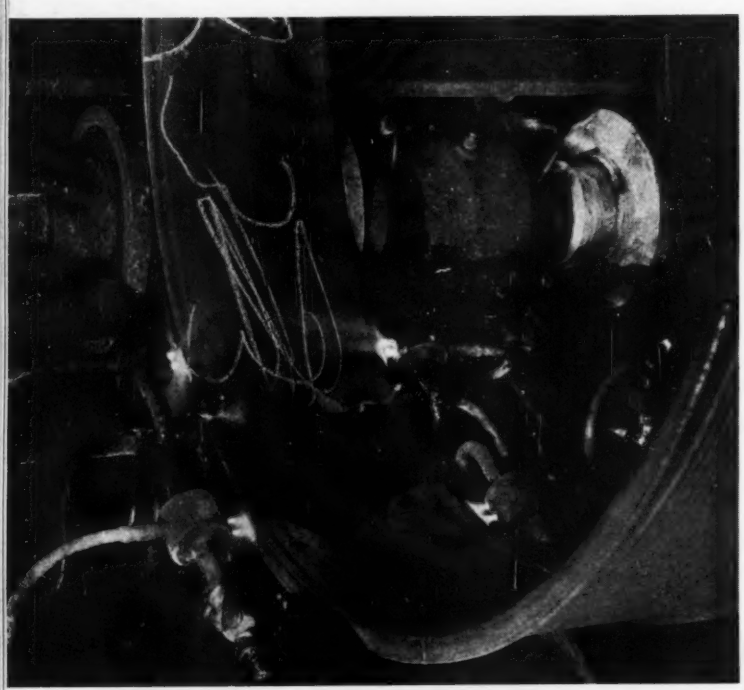


Fig. 3—Close-up of multiple machine shows drive pulleys, hardening and tempering flames, and quenching jet

eration at the Cleveland Welding company on a production basis for heat treating the flanges on the bogie wheel rims. The rims are made of fine-grain modified SAE 1035 steel strip, rolled to the required section at the mill. After cutting to length and cold forming to circular shape, flash welding and cold sizing the diameter, and finally facing the sides to proper width, the rim is ready for flame hardening.

Initial hardness of the rolled plate, as received from the mill, is 150 to 160 brinell. This is increased in the cold forming and sizing operation to 170 to 190 brinell. Specifications require that both flanges of the rim be hardened to between 287 and 360.

The rims are manually mounted, *Fig.* 1, on a cone pressed-air expanding chuck. This chuck is positioned on a sliding table which is moved sideways into operating position by means of a hand lever. When chucked the rim is moved into hardening position and the torches are lighted. An electric motor rotates the chuck at the proper speed, 8 inches per minute, to produce the desired depth of hardness penetration on the rim flanges. Standard welding torches on each side harden and temper both flanges in a single revolution of the wheel. The water quenching jets are located a short distance behind the heating flames, and a vat directly under the rim catches the water. After one complete revolution the primary flame-heating tips are extinguished, but the rim continues to rotate until the end point is hardened by the quench and passes under the drawing tips.

Four Rims Heat Treated at One Time

A multiple production machine is shown in operation in *Fig.* 2. On this machine the flanges on four large bogie wheel rims are flame hardened and tempered simultaneously. These rims are of a different and larger type than those described previously. This machine setup consists of two shafts on which are located four sets of driving pulleys which support and rotate the rims to be hardened. The shafts are connected by a chain and driven by an electric motor. Rims to be flame hardened are mounted on the flanged pulleys and driven at the proper speed past the flames. Guide rollers above hold the rims in alignment and prevent wobbling.

The heating and tempering tips are of special design, as shown in *Fig.* 3, with one set on each side directed at the flanges. The quenching jet is positioned between the flames.

Changes Effected Without Delaying Production

Whenever changes in material or hardness specifications are made, corresponding changes in gas operating pressure and rotational speed are decided on and the desired results obtained on a few rims, after which the operation proceeds in production under the new conditions. Such changes in operation may be effected in a short time without holding up production.

The new process is susceptible to accurate control, and offers wide flexibility. Any desired depth and degree of hardness can be predetermined, after the initial hardening speed and gas pressure are set. Uniformly hardened parts may be produced on a production basis to close specifications.

In the heat-treating field generally, the simultaneous process holds considerable promise. It is particularly applicable in treating parts which require both hardness and impact resistance. This combination of physical characteristics is needed on parts such as, for example, bloom ing mill table rails and crane rails. Where high degrees of hardness are needed, such as on cams, gear teeth, bearings, etc., the standard flame-hardening process will continue to be used.

MACHINE *Editorial* DESIGN

Filling the Gap!

THOSE readers of MACHINE DESIGN interested in national meetings of engineering societies probably will have missed the announcements of meetings ordinarily published each month in this journal. The "Calendar" was discontinued in February, at the time of going to press with that issue, because of the rapidity with which national meetings were then being canceled in compliance with the request of the government.

Necessary as are the restrictions placed on the holding of meetings at this time due to the overtaxing of transportation and hotel facilities, it is to be regretted that their cancelation eliminates the free interchange of ideas among technical men on the national scale encompassed by engineering society gatherings. That such meetings have in the past rendered valuable aid in the war program is evidenced by the fact that many of the major societies have been accorded special recognition by Army, Navy and Ordnance authorities for their contributions to the war effort.

To bridge the gap that will exist until restrictions can logically be removed, local engineering societies established in most of the larger industrial centers—working preferably in collaboration with the local sections of the national engineering societies—should be encouraged to sponsor meetings of such caliber that they would rate as fitting substitutes for the national meetings. Most, if not all, of the major national societies have local chapters that are sufficiently established and well organized to be effective in such a program.

Meetings of this nature already are being held or scheduled to a limited extent. Now the opportunity presents itself for local engineering organizations to put aside their differences and cooperate as never before to meet the current need. Such collaboration, though imposed by wartime conditions, might well serve to consolidate the efforts of the engineering profession as a whole in the present crisis, as well as in the years beyond.

L. E. Jermy

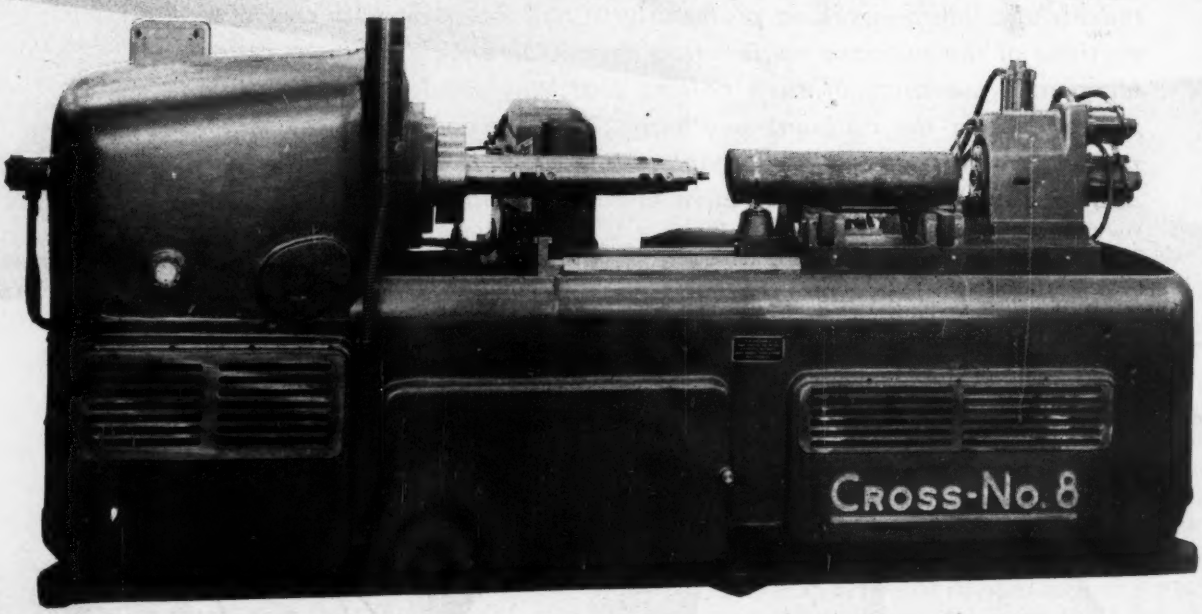
Outstanding Designs—

Shell Centering and Cutoff Machine

To permit the handling of 130-pound shell forgings by inexperienced female help, this machine, developed by The Cross Co., combines an automatic operating cycle with easy loading facilities. Basically it consists of a one-piece, box-type bed on which are mounted the headstock, loading slide and cutoff tool slide.

Carried on the headstock is an SAE 1045 forged steel spindle mounted on hardened, tapered-roller bearings that permit take-up or adjustment. The spindle is driven by worm gearing which is powered by a V-belt drive off the motor. All moving parts have forced-feed lubrication utilizing a vane-type pump. The spindle mounts an expanding mandrel which grips the shell forging from the inside after loading.

Mounted on the loading slide is the housing containing the center drill quill. It has hydraulic horizontal feed and traverse. Loading is accomplished by rolling the shell from a conveyor onto the loading slide elevator rails which hydraulically center the shell in relation to the center of the mandrel. The hydraulically actuated slide then moves horizontally on one flat and one V-way toward the headstock, placing the shell around the arbor. Endwise location of the shell is effected by means of a stop button on the mandrel nose. Hydraulic pusher plates carried on the drill housing exert endwise pressure until the mandrel expands to grip the shell. Pressure then releases and the slide continues forward to a predetermined stop which controls the depth of the center drill.



50-KW Electronic Heater

For ease of installation and handling, and for flexibility in adapting it to different types of operation, this new General Electric heater is built in two cubicles of 1/8-inch reinforced sheet steel, the two being bolted together as a unit when installed.

Each of the two heater units utilizes a timer to control heating and quench time. Each is equipped also with a motor-driven spindle for rotation of the part during heating, if required, and an electrically operated valve to actuate a hydraulic lifting device which facilitates handling of such parts as gears into and out of the inductor coil heating position.

One-half of the steel-enclosed equipment includes a floor-mounted, air-cooled, three-phase transformer which steps up voltage to a bank of six rectifier tubes. Also included are the necessary control and accessory items such as incoming line contactor, transformer tap-changing switch, filament transformers, overload relays, and thyrite discharge resistors—all arranged so that they are readily accessible for inspection or servicing.

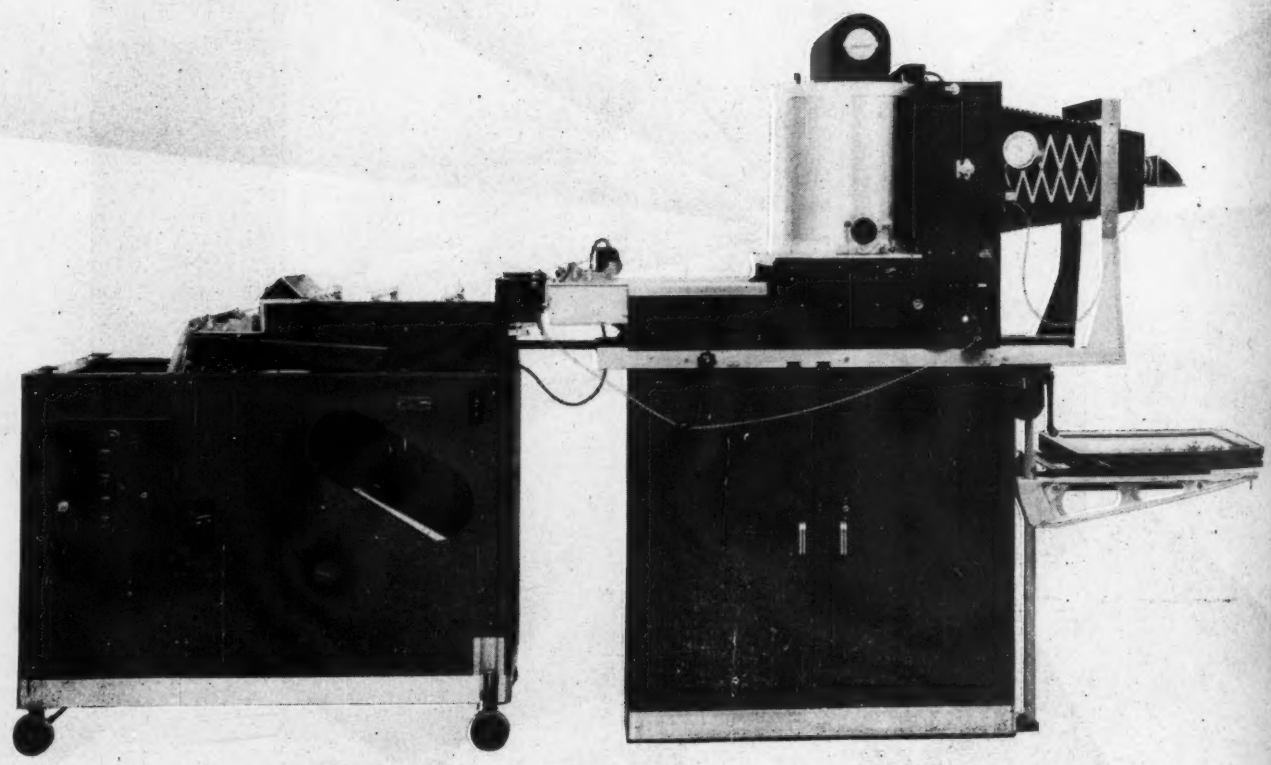
Included in the other half of the heater are a single, large watercooled oscillator tube, a water-flow switch, and a bank of watercooled parallel plate-type capacitors, specially developed for this type of equipment.

Daylight Photocopy Machine

Used for photocopying anything that is written, printed or drawn, these machines, manufactured by The Haloid Co., perform all functions of the complete process including exposure, developing, fixing, washing and drying. The photocopying machine proper is shown at the right. Unit at the left is an automatic washer and dryer.

Mercury-vapor lights, blue-green in color and highly actinic, are used to illuminate the object being copied. In the photocopying machine, the mechanism for removing prints from the developing tank is operated by a solenoid. A covered, spiral-coil developer tank is employed which exposes a minimum surface area of solution to the air. Thus, oxidation and evaporation are minimized. As prints pass into the fixing tray, surplus developer is removed by squeegee rollers.

The washing and drying unit utilizes spray pipes, and a siphon automatically changes the water. A squeegee board is provided for removing excess water from the prints. The double-belt dryer has positive automatic control for both belts. Both machines have all-metal, spot-welded body construction to insure high strength and rigidity. Stainless steel and other corrosion-resistant materials are used for all parts that come into contact with the chemical solutions. Black baking enamel undercoat with black wrinkle finish is used on the housings of the two machines.

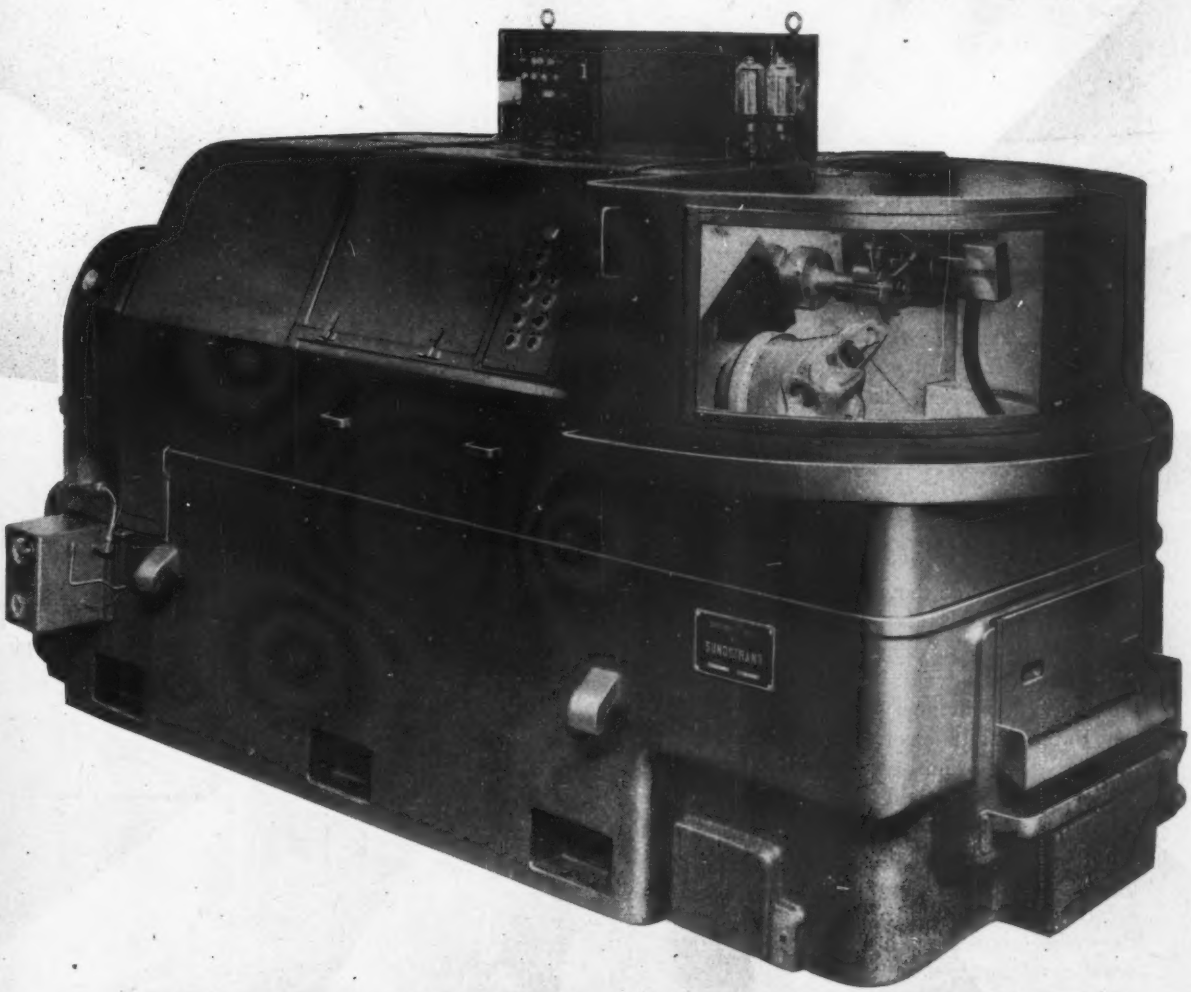


Cylinder Head Fin Miller

With a complete automatic cycle after loading, this machine, designed and manufactured by Sundstrand Machine Tool Co., combines electronic and hydraulic controls. An outstanding feature is the arrangement for controlling the cutter load. The path followed by the cutter is irregular and constantly changing in both shape and depth for each successive fin. With the automatic electronic feed control provided, the fragile cutter is kept loaded to full capacity. If the cut becomes light, the feed rate increases and if the cut becomes heavy, rate of feed decreases.

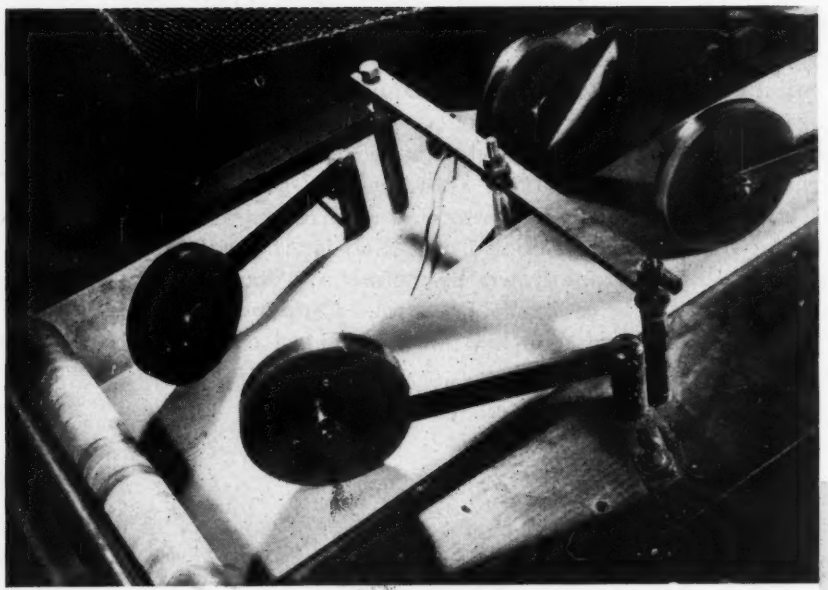
To obtain this variable feed rate the machine is equipped with a 1750/175-revolutions per minute, electronically controlled, adjustable-speed motor for feed drive in place of the conventional constant-speed motor. The spindle-drive motor is of the squirrel-cage induction type and the power required by it is the medium controlled. Since the power of a squirrel-cage induction motor operated between 75 and 125 per cent of its rating is for practical purposes a direct function of a line current drawn by the motor, a current transformer can be used in one of the lines to effect control. Output of the transformer is rectified and amplified, and then fed into the control circuit of the standard electronic control panel.

(*New machines listed on Page* 198)



Applications

of Engineering Parts, Materials and Process



Resists Abrasive Wear

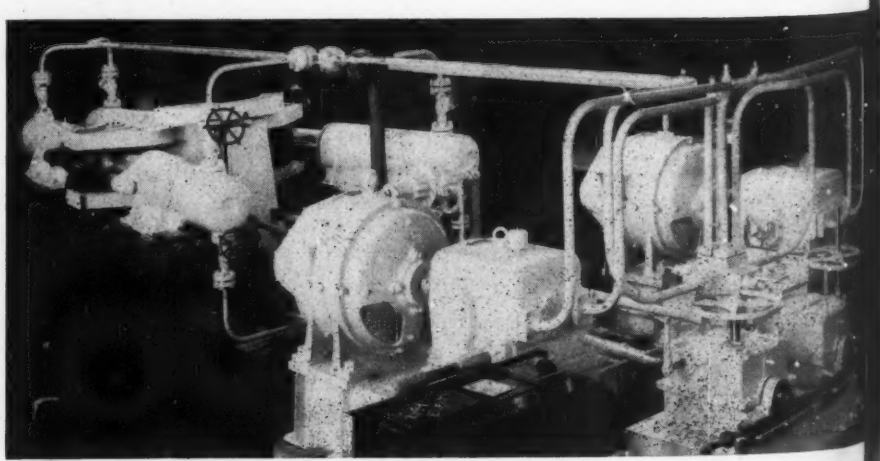
WEAR-RESISTANT disks of Co par, a vinyl resin derivative, a used as shown at left on Wolf pap folding machines to hold down the pap between the feed roller and the foldi mechanism. Abrasive action due to t 15-degree angle of inclination to the d rection of motion resembles that on a automobile tire when the wheels are o of alignment. In this service the Co par disks outlast rubber disks 12 to times, avoiding frequent periodic sto pages for replacement purposes.

Forming Quality, Surface Condition Facilitate Production

VALVE rocker arm cover shown at right is stamped from Armco cold-rolled Paintgrip, the forming qualities of which permit the four-inch draw. The light coating of electrolytic zinc makes it unnecessary to paint the inside, and the mill-bonderized surface allows the exterior to be painted immediately without pretreatment of any kind.

Steering the Victory Ships

HYDRAULICS operates the Victory ship steering gear, which is capable of moving the rudder from hard-over to hard-over (70 degrees) in 30 seconds while the vessel is traveling at 17.78 knots. Shown at right, the Baldwin-built gear is of the Rapson slide electrohydraulic double-ram type. Either of two variable-displacement motor-driven pumps delivers hydraulic power to actuate the rams. Steering is controlled from the wheelhouse by a hydraulic telemotor system which changes the pump stroke to give the required delivery.



Selecting Materials for Corrosion Resistance

By L. G. Vande Bogart
Research Engineer
Crane Co.

DURING a period which has been characterized by an unprecedented demand for metallurgical products, corrosion has done much to aggravate an already critical situation. Because corrosion appears after equipment has consumed precious manufacturing and transportation facilities, arrived at point of usage and been placed in service, it is responsible not only for loss of material but can be and has been responsible for loss of time and lives.

The information presented on the following pages of this Data Sheet is a revision of recommendations which appeared in a recent Crane publication. As a result of the accumulation of additional field data, and with the cooperation of others, it is now possible to include recommendations for a large number of material-fluid combinations not covered in the original charts.

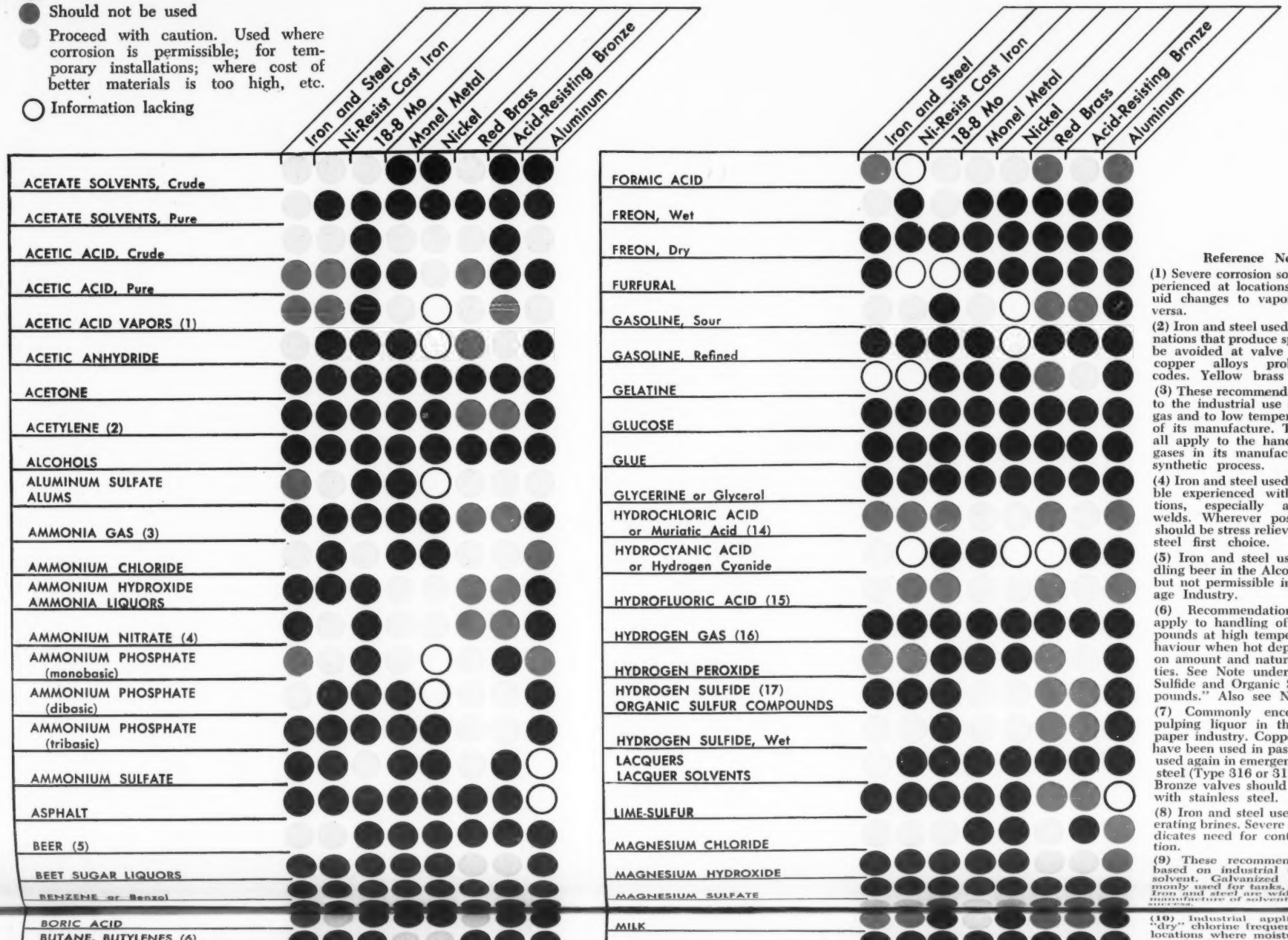
Further, the charts enable prospective users to compare the corrosion resistance of several materials in the same environment in order to determine which could best be used. As a result of current shortages of important alloying elements and of a multitude of manufacturing problems it is not always possible today to obtain the best material for a specific application. Frequently it becomes necessary to select a material on the basis of availability rather than desirability; the charts, therefore, enable engineers to compare an available material with some other material which is known to be better but which is not available.

There already is evidence that lack of familiarity with the accumulated experience of equipment manufacturers and other users has been responsible for some failures and for the resulting need for costly, time-consuming replacements and maintenance.

Although the charts were developed primarily as a guide in the selection of piping material in general and of valve and fitting material in particular, they can be applied to other equipment as well. When one considers that in plants where pumps and other specialized types of equipment are numbered in hundreds, the valves are numbered in thousands or tens of thousands and the length of piping is measured in miles, it at once becomes apparent that equipment of this sort will be exposed to an extremely wide variation of conditions. Experience gained in such service therefore can be a valuable aid to designers of all types of machines and equipment.

KEY TO COLORS

- Can be and is being successfully used
- Should not be used
- Proceed with caution. Used where corrosion is permissible; for temporary installations; where cost of better materials is too high, etc.
- Information lacking



Reference Notes

(1) Severe corrosion sometimes experienced at locations where liquid changes to vapor and vice-versa.

(2) Iron and steel used, but combinations that produce sparks should be avoided at valve seats. High copper alloys prohibited by codes. Yellow brass acceptable.

(3) These recommendations apply to the industrial use of ammonia gas and to low temperature stages of its manufacture. They do not all apply to the handling of hot gases in its manufacture by the synthetic process.

(4) Iron and steel used. Some trouble experienced with hot solutions, especially adjacent to welds. Wherever possible welds should be stress relieved. Stainless steel first choice.

(5) Iron and steel used for handling beer in the Alcohol Industry but not permissible in the Beverage Industry.

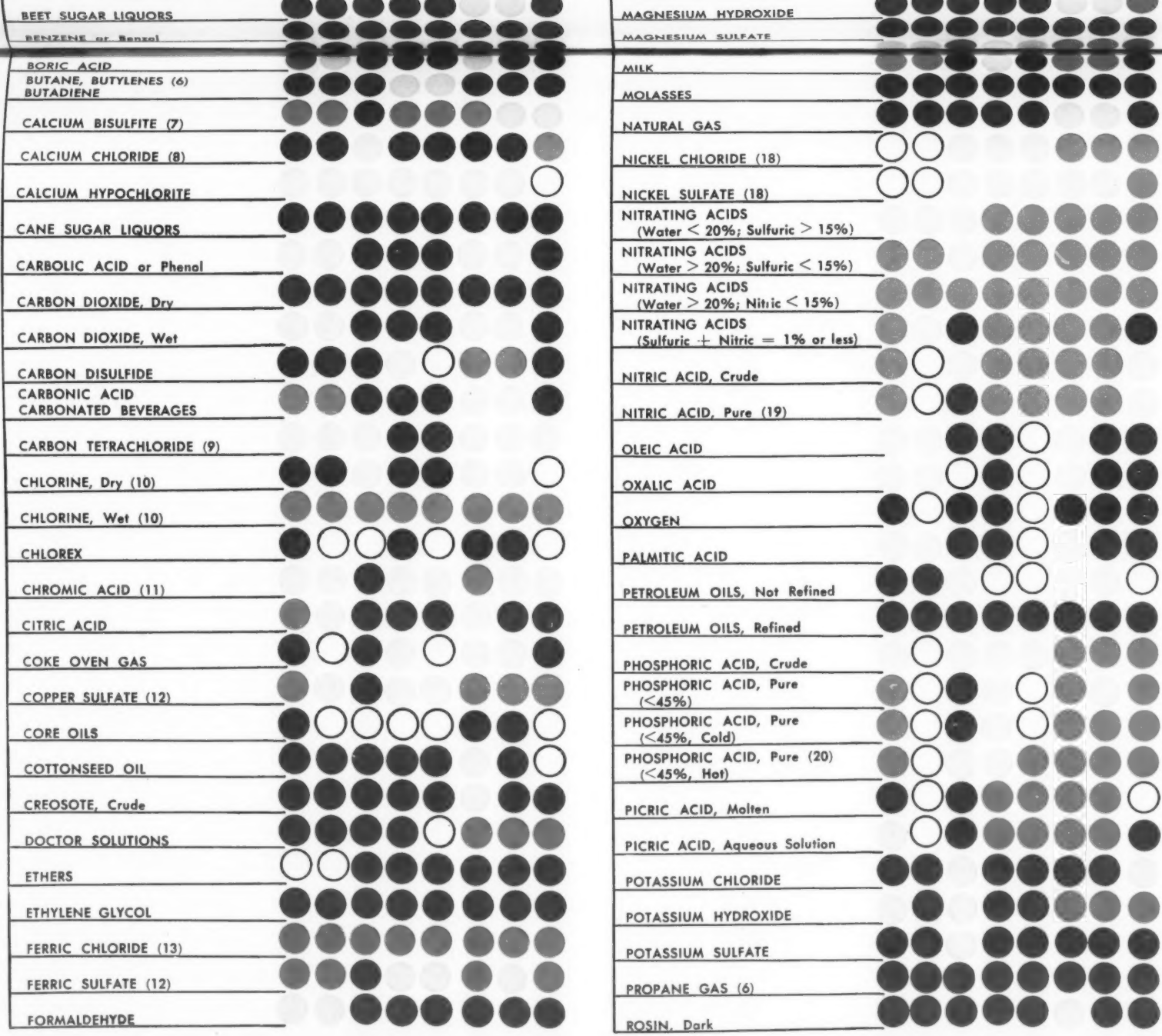
(6) Recommendations do not apply to handling of these compounds at high temperatures. Behaviour when hot depends largely on amount and nature of impurities. See Note under "Hydrogen Sulfide and Organic Sulfur Compounds." Also see Note 16

(7) Commonly encountered as pulping liquor in the pulp and paper industry. Copper and brass have been used in past and can be used again in emergency. Stainless steel (Type 316 or 317) preferred. Bronze valves should be trimmed with stainless steel.

(8) Iron and steel used for refrigerating brines. Severe corrosion indicates need for control of solution.

(9) These recommendations are based on industrial uses of the solvent. Galvanized steel commonly used for tanks, piping, etc. Iron and steel are widely used in manufacture of solvent, with good success.

(10) Industrial applications of "dry" chlorine frequently involve locations where moisture is pres-



(9) These recommendations are based on industrial uses of the solvent. Galvanized steel commonly used for tanks, piping, etc. Iron and steel are widely used in manufacture of solvent, with good success.

(10) Industrial applications of "dry" chlorine frequently involve locations where moisture is present. Steel gives good results as piping material but something better is needed for critical parts. Steel valves trimmed with monel or Hastelloy C give good results. Non-metals required for wet gas and aqueous solutions.

(11) Steel, with or without lead lining, used for tanks, piping, etc. Corrosion of unlined steel adjacent to welds is common. Wherever possible welds should be stress-relieved if steel is not to be lined. Impingement by rapidly moving solution—especially when hot—may cause severe corrosion.

(12) See "Water, Acid Mine." Presence of copper sulfate and/or ferric sulfate in dilute sulfuric acid inhibits the attack of this acid on stainless steel.

(13) Non-metals usually indicated. Hastelloy C used with some success.

(14) Non-metals usually indicated. Hastelloys used with excellent results. Hastelloy B used as trim in steel valves in hydrochloric alkylation. Red brass or bronze valves with nickel alloy trim work well with dilute solutions (<1%). Do not use Hastelloys with acid that has previously been in contact with copper alloys and has become contaminated with copper salts.

(15) Steel standard for piping in hydrofluoric acid manufacture and in hydrofluoric alkylation. Steel valves should be trimmed with monel metal or Hastelloy B.

(16) Recommendations apply to gas when used at ordinary temperatures. Extreme care should be exercised when the gas is to be handled at elevated temperatures, as in hydrogenation and dehydrogenation processes, etc.

(17) Recommendations apply to dry gas at ordinary temperatures. Nickel alloys very susceptible at elevated temperatures.

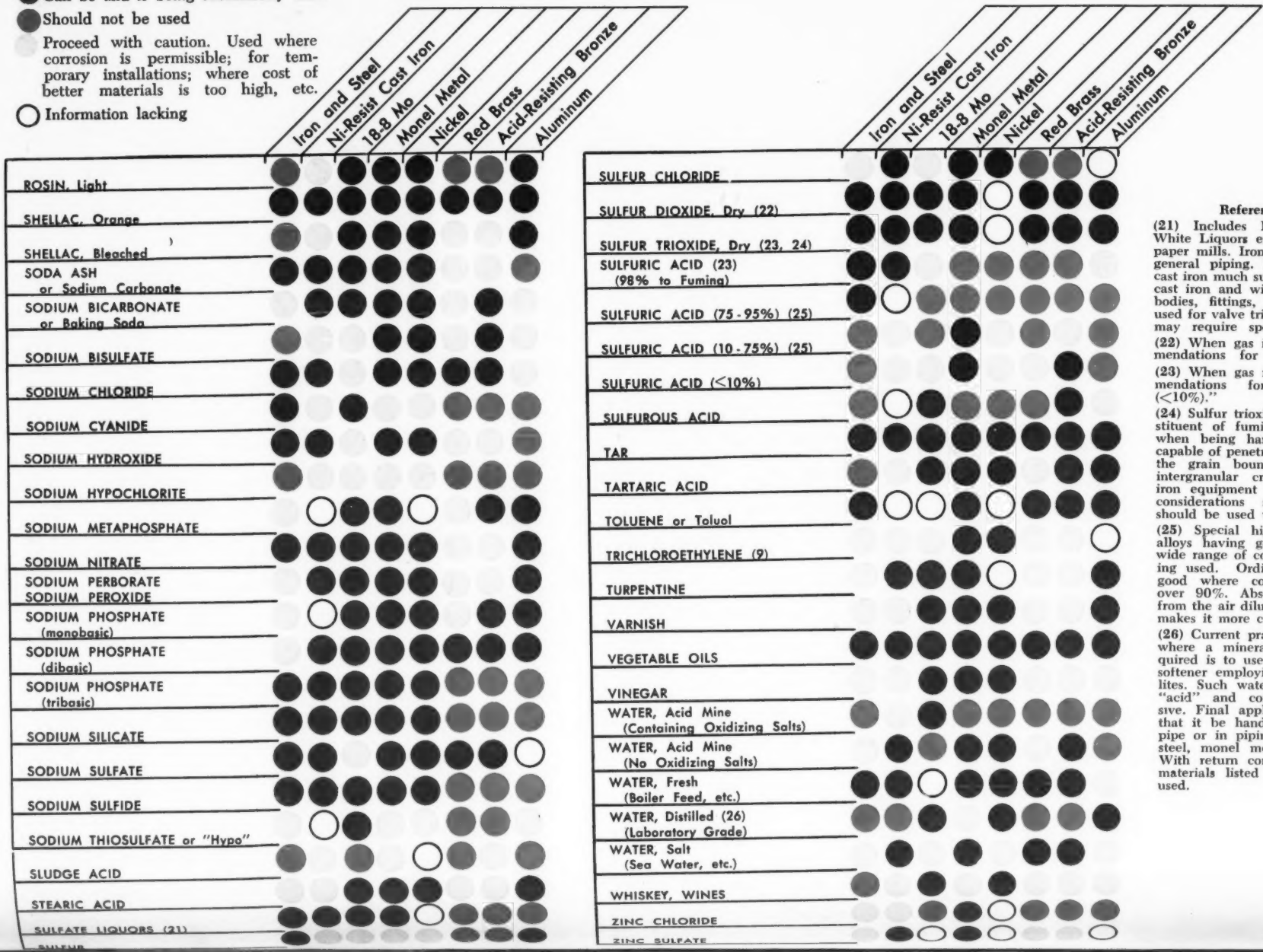
(18) Commonly encountered as constituents of nickel plating solutions, when equipment should be lead, lead-lined, Duriron or rubber-lined.

(19) Straight chromium alloys used in preference to chromium-nickel alloys when feasible.

(20) Special high chromium-nickel alloys required.

KEY TO COLORS

- Can be and is being successfully used
- Should not be used
- Proceed with caution. Used where corrosion is permissible; for temporary installations; where cost of better materials is too high, etc.
- Information lacking



Reference Notes

(21) Includes Black, Green, and White Liquors encountered in Kraft paper mills. Iron and steel used for general piping. Low nickel (1-2%) cast iron much superior to nickel-free cast iron and widely used for valve bodies, fittings, etc. Stainless steel used for valve trim. Evaporator tubes may require special alloys.

(22) When gas is wet apply recommendations for "Sulfurous Acid."

(23) When gas is wet apply recommendations for "Sulfuric Acid (<10%)."

(24) Sulfur trioxide, either as a constituent of fuming sulfuric acid or when being handled as a gas, is capable of penetrating cast iron along the grain boundaries and causing intergranular cracking. Much cast iron equipment is used, but safety considerations suggest that steel should be used wherever possible.

(25) Special high chromium-nickel alloys having good resistance to a wide range of concentrations are being used. Ordinary iron and steel good where concentration remains over 90%. Absorption of moisture from the air dilutes exposed acid and makes it more corrosive to iron.

(26) Current practice in many cases where a mineral free water is required is to use the effluent from a softener employing "hydrogen" zeolites. Such water may be distinctly "acid" and correspondingly corrosive. Final application may require that it be handled in rubber lined pipe or in piping made of stainless steel, monel metal, etc.
With return condensate any of the materials listed can be successfully used.

MACHINE DESIGN'S

MATERIALS WORK SHEET

FILING NUMBER
21.00

Synthetic Rubbers

Five basic types of synthetic rubber serve in this country's war program and undoubtedly will offer much in postwar design. They are: Buna-S, Buna-N, Neoprene*, Butyl and Thiokol*. Hundreds of compounds are produced of each of these basic types, each compound being developed to meet specific engineering requirements. For the sake of clarity and brevity, comparative properties must be confined at the present to indicative terms rather than definite figures.

To aid further in the selection of synthetic rubbers, charts and data abstracted from tentative specifications of the American Society for Testing Materials (ASTM No. D735-43T)are given.

*Tradename

AVAILABLE IN:

The crude synthetic rubbers are available as latex or in sheets (see table of Comparative Properties). Generally, synthetic rubbers can be obtained in standard forms such as tubes, sheets, moldings, rubberized fabrics, cements and adhesives, sponge materials, etc.

COMPARATIVE PROPERTIES§

	Natural Rubber	Buna S Type	Buna N Types	Neoprene Types	Butyl Type	Thiokol Polysulfide Types
Properties Important in Processing						
Form in which available	Latex and Solid	Latex and Solid	Latex and Solid	Latex and Solid	Solid Form	Dispersion, Solid & Powder
Breakdown	Very Good	Good	Fair	Good—V.G.	None	Slow
Plasticity Range after Breakdown	High—Low	High—Low	High—Medium	High—Low	Medium	High—Low
Building Tack and Cohesion	Excellent	Fair	Fair	Very Good	Good	Fair—Good
Vulcanizability	Very Good	Very Good	Very Good	Very Good	Fair—Good	Fair
General Processability	Very Good	Good	Fair	Good	Fair	Fair
Properties Important in Application						
Extensibility	Excellent	Good	Good	Excellent	Excellent	Good
Resilience	Excellent	Good	Fair—Good	Very Good	Low	Good
Tensile	Excellent	Fair—Good	Good	Very Good	Good	Fair
Electrical Properties	Excellent	Excellent	Fair	Fair	Excellent	Fair
Impermeability to Gases	Good	Good	Good	Very Good	Excellent	Excellent
Impermeability to Water	Good—V.G.	Fair—Good	Fair—Good	Fair—Good	Very Good	Very Good
Resistance to: Plastic Flow	Very Good	Good	Good	Good	Fair—Good	Low
Abrasion	Very Good	Good—V.G.	Good—V.G.	Very Good	Fair	Low
Tear	Very Good	Fair—Good	Fair—Good	Good	Fair—V.G.	Fair—Good
Heat	Good	Fair—V.G.	Fair—V.G.	Very Good	Fair	Low
Cold	Very Good	Very Good	Fair—Good	Fair—V.G.	Good	Fair—Good
Resistance to: Air	Fair	Good	Good	Excellent	Excellent	Excellent
Ozone	Inadequate	Inadequate	Fair	Excellent	Excellent	Excellent
Light	Fair	Fair	Low	Excellent	Excellent	Excellent
Petroleum	Low	Low	Excellent	Good	Low	Excellent
Aromatic Oils	Inadequate	Inadequate	Fair	Low	Inadequate	Excellent

Note: This chart is not intended to be used for the specification of materials. It gives only general indications of the inherent qualities of the basic types listed. Final specifications for individual applications should be worked out in conjunction with a producer of synthetic rubbers.

§From *The Five Commercial Types of Synthetic Rubber*, published by United States Rubber Co.

MACHINE DESIGN is pleased to acknowledge the collaboration of the following companies in this presentation: E. I. DuPont DeNemours & Co.; The Firestone Tire & Rubber Co.; The B. F. Goodrich Co.; The Goodyear Tire & Rubber Co.; Hycar Chemical Co.; Stanco Distributors Inc.; Thiokol Corp.; United States Rubber Co.

USE OF TABLES I, II, III AND IV

These tables list the specifications pertaining to synthetic rubbers set up jointly by committees of the Society of Automotive Engineers and the American Society for Testing Materials in 1943. The specifications cover compounds designated by the prefix letters *R* and *S*. Compounds carrying the prefix *R* are for services where no resistance to oil is required. Those carrying the prefix *S* are for services where resistance to the action of petroleum-base oils or other organic fluids is required.

CLASSES: Type *R* compounds are divided into two classes; *RN* and *RS*. Class *RN* will not be dealt with here inasmuch as it pertains only to compounds of natural rubber. Class *RS* compounds (see TABLE I) contain synthetic rubber or rubber-like materials to the extent of 50 per cent or more by volume of the total elastomer content.

Type *S* compounds, made of synthetic rubber or rubber-like materials, are divided into three classes designated by the letters *A*, *B* and *C* added to the *S* prefix. Class *SA* indicates compounds having maximum oil resistance, Class *SB* applies to compounds of good oil resistance, and Class *SC* applies to compounds which offer what is termed medium oil resistance.

GRADES: Each class of compound has a number of different grades, each having different physical properties. Grades are designated by numbers following the prefix letters and, when necessary by suffix letters after the grade number. Grade numbers consist of three digits the first indicating the durometer hardness range; for example, 3 for 30 plus or minus 5, 4 for 40 plus or minus 5, etc. The next two digits indicate the minimum tensile strength; for example, 09 for 900 pounds per square inch, 15 for 1500 pounds per square inch, etc.

SUFFIX NUMBERS: Used only in cases where compounds having the same tensile strength require different ultimate elongations. They are single digits preceded by a hyphen and are appended to the grade numbers, indicating the required minimum elongations; for example, —4 for 400 per cent, —7 for 700 per cent, etc.

TABLE I

Physical Requirements of Rubber Compounds

Type R, Class RS (Non-oil Resistant)

	Basic Requirements			Requirements Added by Suffix Letter Suffix A Oven Aged 70 Hrs. at 158 F.		Suffix B
Grade Number	Durometer Hardness Number	Tensile Strength (min psi)	Ultimate Elongation (min %)	Change in Tensile Strength (max %)	Change in Ultimate Elongation (max %)	Compression Set (max %)
RS300	30 ± 5	....	....	....	....	25
RS307	30 ± 5	700	300	—25	—25	50
RS309	30 ± 5	900	300	—25	—25	50
RS309-5	30 ± 5	900	500	—25	—25	50
RS315	30 ± 5	1500	600	—25	—25	50
RS320	30 ± 5	2000	500	—25	—25	50
RS320-7	30 ± 5	2000	700	—25	—25	50
RS400	40 ± 5	....	....	....	....	25
RS402	40 ± 5	200	150	—25	—25	50
RS405	40 ± 5	500	300	—25	—25	50
RS405-7	40 ± 5	500	700	—25	—25	50
RS409	40 ± 5	900	400	—25	—25	50
RS415	40 ± 5	1500	500	—25	—25	50
RS415-7	40 ± 5	1500	700	—25	—25	50
RS420	40 ± 5	2000	700	—25	—25	50
RS425	40 ± 5	2500	600	—25	—25	50
RS500	50 ± 5	....	....	....	....	25
RS503	50 ± 5	300	125	—25	—25	50
RS506	50 ± 5	600	300	—25	—25	50
RS509	50 ± 5	900	300	—25	—25	50
RS509-5	50 ± 5	900	500	—25	—25	50
RS515	50 ± 5	1500	400	—25	—25	50
RS515-6	50 ± 5	1500	600	—25	—25	50
RS518	50 ± 5	1800	400	—25	—25	50
RS520	50 ± 5	2000	600	—25	—25	50
RS600	60 ± 5	....	....	....	....	25
RS604	60 ± 5	400	100	—25	—25	50
RS606	60 ± 5	600	300	—25	—25	50
RS609	60 ± 5	900	300	—25	—25	50
RS612	60 ± 5	1200	500	—25	—25	50
RS615	60 ± 5	1500	350	—25	—25	50
RS620	60 ± 5	2000	600	—25	—25	50
RS625	60 ± 5	2500	450	—25	—25	50
RS700	70 ± 5	....	....	....	....	25
RS703	70 ± 5	300	100	—25	—25	50
RS704	70 ± 5	400	150	—25	—25	50
RS706	70 ± 5	600	200	—25	—25	50
RS709	70 ± 5	900	250	—25	—25	50
RS709-4	70 ± 5	900	400	—25	—25	50
RS715	70 ± 5	1500	300	—25	—25	50
RS715-5	70 ± 5	1500	500	—25	—25	50
RS720	70 ± 5	2000	350	—25	—25	50
RS725	70 ± 5	2500	300	—25	—25	50
RS804	80 ± 5	400	50	—25	—25	50
RS806	80 ± 5	600	150	—25	—25	50
RS809	80 ± 5	900	200	—25	—25	50
RS812	80 ± 5	1200	250	—25	—25	50
RS820	80 ± 5	2000	200	—25	—25	50
RS825	80 ± 5	2500	300	—25	—25	50
RS907	90 ± 5	700	75	—25	—25	..
RS915	90 ± 5	1500	75	—25	—25	..

SUFFIX LETTERS: May be added singly or in combination after any grade number to indicate additional requirements beyond those specified in the tables. The significance of the approved suffix letters is as indicated in the following listing:

A . . . . . Oven aging test required
B . . . . . Compression set test required
C . . . . . Weather resistance required
D . . . . . Load deflection test required
E . . . . . Oil aging test required
F . . . . . Flexible at —40 F. (S.A.E.-A.S.T.M. Test)
FF . . . . . Flexible at —70 F.(S.A.E.-A.S.T.M. Test)

Example: Grade *SB* 625 F designates a synthetic compound with good oil resistance, having a durometer hardness of 60 plus or minus 5, and an original minimum tensile strength of 2500 pounds per square inch. It must pass the low-temperature brittleness test at —40 degrees Fahr. and conform to the other basic requirements prescribed in TABLE III.

CHARACTERISTICS

BUNA S TYPES: Co-polymers of butadiene, C_4H_6 and styrene, $C_2H_3(C_6H_5)$. Selected for the major part of the government's synthetic program. Of all synthetics, they are the most similar to rubber in processing and performance characteristics. Can be vulcanized with sulphur and rubber accelerators and cured to hard rubber. Resistance to deterioration in air both at room temperature and elevated temperatures is considerably better than natural rubber. In most applications may be used to replace rubber. To bring out best physical properties, buna S must be compounded with a black pigment, properties of the pure gum stock being inferior. Buna S is known also as GR-S (Government Rubber S).

BUNA N TYPES: These are co-polymers of butadiene and acrylonitrile. The N in the name refers to the nitrile radical. Buna N's offer excellent resistance to oil. Similar to rubber in that they are vulcanized with sulphur and rubber accelerators and may be cured to hard rubber. However, they

TABLE II

Physical Requirements of Synthetic Rubber Compounds
Type S, Class SA (Maximum Oil Resistance)

Basic Requirements						Suffix E†	
					Oil Aged 70 Hrs. at 158 F	Oil Aged 70 Hrs. at 158 F	
Grade Number	Durometer Hardness Number	ASTM Hardness Number**	Tensile Strength (min psi)	Ultimate Elongation (min %)	% Change in Volume (limits)	Change in Tensile Strength (max %)	Change in Durometer Hardness (limits)
SA-302	30 ± 5	150 ± 20	200	500	—15 to +5	—20	—5 to +10
SA-403	40 ± 5	105 ± 15	300	500	—5 to +3	—20	—5 to + 5
SA-506	50 ± 5	75 ± 10	600	500	—3 to +3	—20	—5 to 0
SA-607	60 ± 5	55 ± 8	700	400	—3 to +3	—20	—5 to 0
SA-706	70 ± 5	40 ± 7	600	400	—3 to +3	—20	—5 to 0
SA-709	70 ± 5	40 ± 7	900	500	—3 to +3	—20	—5 to 0
SA-809	80 ± 5	28 ± 7	900	200	—3 to +3	—20	—5 to 0
SA-910	90 ± 5	13 ± 7	1000	100	—3 to +3	—20	—5 to 0

**Required for referee purposes only.
†Requirement additional to the basic requirements. It is specified by adding the suffix *E* to the grade number. Suffixes *A* and *B*, indicating oven aging and compression set respectively, are omitted for Class *SA* compounds because these materials may be thermoplastic.

TABLE III

Physical Requirements of Synthetic Rubber Compounds
Type S, Class SB (Good Oil Resistance)

Basic Requirements						Requirements Added by Suffix Letters					
						Suffix E Oil Aged 70 Hrs. at 212 F		Suffix A Oven Aged 70 Hrs. at 212 F			Suffix B Compression Set (max %)
Grade Number	Durometer Hdns. Number	ASTM Hdns. No.**	Tensile Strength (min psi)	Ultimate Elong. (min %)	Oil Aged 70 Hrs. at 212 F % Change in Volume (limits)	Change in Tensile Strength (max %)	Change in Duro. Hdns. (limits)	Change in Tensile Strength (max %)	Change in Ult. Elong. (max %)	Change in Duro. Hdns. (limits)	22 Hrs. at 158 F	70 Hrs. at 212 F
SB-305	30 ± 5	150 ± 20	500	400	—15 to +25	—20	—10 to +20	—20	—50	0 to +20	50	..
SB-309	30 ± 5	150 ± 20	900	500	—15 to +25	—20	—10 to +20	—20	—50	0 to +20	50	..
SB-400	40 ± 5	105 ± 15	...	...	—15 to +25	...	—10 to +15	..	..	0 to +15	..	70
SB-405	40 ± 5	105 ± 15	500	400	—15 to +25	—20	—10 to +20	—20	—50	0 to +20	45	..
SB-410	40 ± 5	105 ± 15	1000	500	—15 to +25	—20	—10 to +20	—20	—50	0 to +20	45	..
SB-415	40 ± 5	105 ± 15	1500	500	—15 to +25	—20	—10 to +15	—20	—50	0 to +15	45	..
SB-500	50 ± 5	75 ± 10	...	...	—5 to +25	...	—10 to +10	...	...	0 to +10	..	65
SB-508	50 ± 5	75 ± 10	800	400	—5 to +25	—20	—10 to +15	—20	—50	0 to +15	40	..
SB-515	50 ± 5	75 ± 10	1500	500	—5 to +25	—20	—10 to +15	—20	—50	0 to +15	40	..
SB-520	50 ± 5	75 ± 10	2000	500	—5 to +25	—20	—10 to +10	—20	—50	0 to +10	40	..
SB-600	60 ± 5	55 ± 8	...	...	—5 to +25	...	—10 to +10	...	...	0 to +10	..	65
SB-608	60 ± 5	55 ± 8	800	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +15	35	..
SB-612	60 ± 5	55 ± 8	1200	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +15	35	..
SB-617	60 ± 5	55 ± 8	1700	400	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	35	..
SB-625	60 ± 5	55 ± 8	2500	500	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	35	..
SB-700	70 ± 5	40 ± 7	...	...	—5 to +25	...	—10 to +10	...	...	0 to +10	..	65
SB-710	70 ± 5	40 ± 7	1000	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-715	70 ± 5	40 ± 7	1500	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-720	70 ± 5	40 ± 7	2000	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-725	70 ± 5	40 ± 7	2500	300	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-730	70 ± 5	40 ± 7	3000	400	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-800	80 ± 5	28 ± 7	...	...	—5 to +25	...	—10 to +5	...	...	0 to +10	..	65
SB-804	80 ± 5	28 ± 7	400	100	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-810	80 ± 5	28 ± 7	1000	150	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-815	80 ± 5	28 ± 7	1500	150	—5 to +25	—20	—10 to +10	—20	—60	0 to +10	30	..
SB-820	80 ± 5	28 ± 7	2000	150	—5 to +25	—20	—10 to +5	—20	—60	0 to +10	30	..
SB-902	90 ± 5	13 ± 7	200	50	—5 to +25	—20	—10 to +5	—20	—60	0 to +5	40	..
SB-910	90 ± 5	13 ± 7	1000	100	—5 to +25	—20	—10 to +5	—20	—60	0 to +5	30	..

**Required for referee purposes only.

differ from rubber in other processing characteristics and in some operations are more difficult to handle. Buna N's have good "breakaway" properties when in contact with metals. In other words when used, for example, for seals on a hydraulic piston, there is little tendency for the rubber to stick to the metal cylinder.

NEOPRENE TYPES: Polymers of chloroprene, C_4H_5Cl. Were our first commercially successful synthetic rubbers. Good general-purpose rubbers, they offer good resistance to chemicals and oils, and excellent resistance to heat, air and light. In fact they have better light resistance than any other rubber. Another important quality is their excellent flame resistance. For this reason they are used as an outer sheath for electrical insulation even though the electrical properties are lower than those of natural rubber. Neoprene type GN carries government designation GR-M.

BUTYL TYPES: Co-polymers of isobutylene, C_4H_8, and small amounts of other unsaturated hydrocarbons such as butadiene, C_4H_6, or isoprene, C_5H_8. They have processing characteristics comparable to natural rubber but require a longer curing time. Cannot be vulcanized to hard rubber. General resistance to deterioration is good, resistance to oxygen and acid very good, but physical properties are lower than those of natural rubber. Offer excellent impermeability to gases and partly because of this they are resistant to mustard gas and are one of the few materials resistant to Lewisite. Used advantageously in applications where resistance to chemicals and oxidation are more important factors than properties such as tensile strength and resistance to cold flow. Government designation is GR-1.

THIOKOL POLYSULFIDE TYPES: Types A, FA, and ST are made from dihalogenated organic compounds and inorganic polysulfide or combinations and modifications of these. In general, these synthetic rubbers show low swell and excellent resistance to deterioration in oils, greases, solvents and lacquer thinners. They have better resistance to aromatic hydrocarbons and aromatic blended gasolines than natural rubber or other synthetic rubbers. They also show excellent resistance to ozone, sun light and natural aging. Their tensile strengths, while not as high as natural rubber, are adequate for many mechanical rubber goods compounds. Stocks that show good resilience, elongation, low-temperature flexibility, abrasion resistance and resistance to cold flow are readily obtainable. Thiokol synthetic rubbers are used to advantage in applications where low swell, good solvent resistance, and good aging characteristics are desired. Government designation is GR-P.

APPLICATIONS

BUNA S TYPES: Having excellent electrical properties, they are used widely for electric cable insulation. Good for seals and gaskets for water pumps, but not for oil equipment.

BUNA N TYPES: Oil and gasoline hose, tank lining, packings, gaskets, printers' blankets, rubber-covered rolls, valve inserts, molded dust seals, diaphragms for hydraulic gear shifts, bushing seals, piston seals, bullet-sealing hose, hydraulic lines, hydraulic system boots, hydraulic packing seals, stoker bellows, motor cushion rings, oil-resisting tires, vibration dampeners and special molded shapes to specifications.

NEOPRENE TYPES: Oil-resistant hose, belting, gaskets, sheet goods, tubes, tape, fabrics, latex tank-crash padding, fuel cells, cements, sheet packing, printing rollers, textile rolls, solid truck tires, airplane tires, electric cable covering, airplane wing and hull caulking, vibration dampeners, diaphragms, coatings for metal tanks, pipe linings, and numerous special shapes which can be molded to specifications.

TABLE IV
Physical Requirements of Synthetic Rubber Compounds
Type S, Class SC (Medium Oil Resistance)

	Basic Requirements					Requirements Added by Suffix Letters				
						Suffix E Oil Aged 70 Hrs. at 212 F		Suffix A Oven Aged 70 Hrs. at 212 F			Suffix B Compression Set (max %)	
Grade Number	Durometer Hdns. Number	ASTM Hdns. No.**	Tensile Strength (min psi)	Ultimate Elong. (min %)	Oil Aged 70 Hrs. at 212 F % Change in Volume (limits)	Change in Tensile Strength (max %)	Change in Duro. Hdns. (limits)	Change in Tensile Strength (max %)	Change in Ult. Elong. (max %)	Change in Duro. Hdns. (limits)	22 Hrs. at 158 F	70 Hrs. at 212 F
SC-305	30 ± 5	150 ± 20	500	500	100 to 150	—85	—30 to —10	—15	—40	0 to +15	70	
SC-310	30 ± 5	150 ± 20	1000	500	100 to 150	—85	—30 to —10	—15	—40	0 to +15	70	
SC-315	30 ± 5	150 ± 20	1500	500	100 to 150	—85	—30 to —10	—15	—40	0 to +15	70	
SC-320	30 ± 5	150 ± 20	2000	500	100 to 150	—85	—30 to —10	—15	—40	0 to +15	70	
SC-400	40 ± 5	105 ± 15	....	....	90 to 140	....	—35 to —15	....	....	0 to +15	..	
SC-405	40 ± 5	105 ± 15	500	500	90 to 140	—80	—35 to —15	—15	—40	0 to +15	65	
SC-415	40 ± 5	105 ± 15	1500	500	90 to 140	—80	—35 to —15	—15	—40	0 to +15	65	
SC-420	40 ± 5	105 ± 15	2000	500	90 to 140	—80	—35 to —15	—15	—40	0 to +15	65	
SC-425	40 ± 5	105 ± 15	2500	500	90 to 140	—80	—35 to —15	—15	—40	0 to +15	65	
SC-500	50 ± 5	75 ± 10	....	....	80 to 130	....	—40 to —20	....	....	0 to +15	..	
SC-507	50 ± 5	75 ± 10	700	300	80 to 130	—70	—40 to —20	—15	—40	0 to +15	60	
SC-512	50 ± 5	75 ± 10	1200	300	80 to 130	—70	—40 to —20	—15	—40	0 to +15	60	
SC-520	50 ± 5	75 ± 10	2000	400	80 to 130	—70	—40 to —20	—15	—40	0 to +15	60	
SC-525	50 ± 5	75 ± 10	2500	400	80 to 130	—70	—40 to —20	—15	—40	0 to +15	60	
SC-530	50 ± 5	75 ± 10	3000	500	80 to 130	—70	—40 to —20	—15	—40	0 to +15	60	
SC-600	60 ± 5	55 ± 8	....	....	70 to 120	....	—50 to —20	....	....	0 to +10	..	
SC-608	60 ± 5	55 ± 8	800	300	70 to 120	—60	—50 to —20	—15	—40	0 to +15	55	
SC-614	60 ± 5	55 ± 8	1400	300	70 to 120	—60	—50 to —20	—15	—40	0 to +10	55	
SC-620	60 ± 5	55 ± 8	2000	300	70 to 120	—60	—50 to —20	—15	—40	0 to +10	55	
SC-625	60 ± 5	55 ± 8	2500	300	70 to 120	—60	—50 to —20	—15	—40	0 to +10	55	
SC-630	60 ± 5	55 ± 8	3000	400	70 to 120	—60	—50 to —20	—15	—40	0 to +10	55	
SC-700	70 ± 5	40 ± 7	....	....	60 to 110	....	—50 to —25	....	....	0 to +10	..	
SC-707	70 ± 5	40 ± 7	700	200	60 to 110	—60	—50 to —25	—15	—40	0 to +10	50	
SC-717	70 ± 5	40 ± 7	1700	200	60 to 110	—60	—50 to —25	—15	—40	0 to +10	50	
SC-725	70 ± 5	40 ± 7	2500	300	60 to 110	—60	—50 to —25	—15	—40	0 to +10	50	
SC-800	80 ± 5	28 ± 7	....	....	50 to 100	—40	—50 to —30	....	....	0 to +10	..	
SC-804	80 ± 5	28 ± 7	400	100	50 to 100	—40	—50 to —30	—15	—40	0 to +10	50	
SC-808	80 ± 5	28 ± 7	800	100	50 to 100	—40	—50 to —30	—15	—40	0 to +10	50	
SC-815	80 ± 5	28 ± 7	1500	100	50 to 100	—40	—50 to —30	—15	—40	0 to +10	50	
SC-902	90 ± 5	13 ± 7	200	50	40 to 90	—30	—50 to —30	—15	—40	0 to +10	50	
SC-910	90 ± 5	13 ± 7	1000	100	40 to 90	—30	—50 to —30	—15	—40	0 to +10	50	

**Required for referee purposes only.

BUTYL TYPES: Most of this synthetic rubber is used today for the production of Ordnance inner tubes. It serves splendidly in all types of acid-handling equipment. Advantages of butyl in inner tubes are: It holds air three times better than natural rubber, has good tear resistance, and has the ability to envelope small foreign particles that may be present in the compound.

THIOKOL TYPES: Used where resistance to deterioration is more important than physical properties. Typical applications are paint spray and solvent hose, certain packings and gaskets, oil suction and discharge hose, printers' rolls, tank linings, cable covers, tubing, sheet goods, tape, solvent-resisting sealing putty, selector valve rings, diaphragms, and cements. In general, excellent for all oil solvent, ozone, and sunlight resistant applications. Particularly effective in connection with aromatic blended fuels and lacquer solvents.

SELECTING AND SPECIFYING

Where the data given in this work sheet is not sufficiently detailed to permit reliable selection of a synthetic rubber for a specific application, it is suggested that the designer contact a producer for recommendations.

The producer should be supplied with information such as: Where the part is to be used and under what operating conditions? What temperature range it must perform in? If oil or solvent will contact the part, what type of oil is involved—paraffinic or aromatic? Also, will the exposure be intermittent or continuous? Must part operate in presence of water? Of ozone? What type or types of loading and of what magnitudes will they be? Will loads be applied intermittently or continuously? Will part be subject to flexing? How much? To abrasion? What electrical properties, if any, are required?

SUITABILITY CHART

	Natural Rubber	Neoprene	Thiokol	Buna N	Buna S	Butyl
Acetone	C	B	C	A	C	C
Acids	C‡	C‡	B	C‡	C‡	C
Alcohols						
Ethyl	C	C	C	C	C	C
Amyl	C	C	C	C	C	C
Benzyl		A	B	A		C
Alkalies	C	C	C	C	C	C
Animal Oils & Fats	B	C	C	C	B	C
Benzene	A	A	B	B	A	B
Butyl Acetate	A	A	B	A	A	C
Butane	A	C	C	C	A	A
Carbon Disulphide	A	A	A	B	A	A
Carbon Tetrachloride	A	A	C	C	A	A
Chloroform	A	A	B	A	A	A
Cottonseed Oil	B	C	C	C	B	C
Cresol	A	A	B	B	A	C
Dibutyl Phthalate			B	A		A
Diethylene Glycol		C	C	C		C
Dioctyl Phthalate			B	C		A
Ethyl Acetate	A	A	C	A	A	C
Ethylene Glycol	C	C	C	C	C	C
Furfural	A	A	A	A	A	C
Gasoline	A	B	C	C	A	A
Glycerine	C	C	C	C	C	C
Heptane	A	B	C	C	A	A
Hexane	A	B	C	C	A	A
Kerosene	A	B	C	C	A	A
Mineral Oils	A	B	C	C	A	A
Naphtha, Petroleum	A	B	C	C	A	A
Nitro Benzene	A	A	A	A	A	A
Octane	A	B	C	C	A	A
Oils, Mineral	A	B	C	C	A	A
Oils, Vegetable	B	C	C	C	B	C
Ozone	Not Suit.	Good Res.	Good Res.	Not Suit.	Not Suit.	Good Res.
Phenol	A	A	B	B	A	C
Propane	A	B	C	C	A	A
Toluene	A	A	B	B	A	B
Turpentine	A	B	B	C	A	A
Xylene	A	A	B	B	A	B

A—Generally unsuitable; swells or softens.
B—Suitable under some conditions; swelling or softening is moderate.
C—Is especially suitable; little or no swelling or softening.
‡Except strong oxidizing acids.
Note: Suitability ratings apply to representative compounds and are intended to be indicative rather than specific.

PROFESSIONAL VIEWPOINTS

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

". . . supports research"

To the Editor:

I am glad to see from your fine editorial in the January issue of MACHINE DESIGN that you are giving your full support to the prosecution of research as a major preparedness effort during the next peace period. Research is in my opinion, of such importance that it merits the greatest possible attention in the postwar planning of the executive departments of the Government, as well as of Congress.

I have looked over your magazine with great interest and want to congratulate you on the high and interesting quality of the articles as well as the attractive presentation of your advertisements.

—J. A. FURER, *Rear Admiral, U.S.N. (Ret.)*
Coordinator of Research and Development
Navy Department

". . . cuts were transposed"

To the Editor:

In your abstract of the writer's recent S.A.E. paper (MACHINE DESIGN, January 1945, Pages 147 to 152) the word "Zerol" has been replaced by "curved tooth" in several instances. While it is appreciated that this may have been done to avoid excessive repetition of the same term, and with the thought that the context made quite clear what type of gear was referred to in each case, we feel that the result is apt to be misleading. This is because Zerols, spirals and hypoids are all curved-tooth gears, each with its individual characteristics. Accordingly, the words "curved-tooth" should be replaced by "Zerol" in the following places: Page 148, Col. 1, lines 4 and 17; Page 149, Col. 1, Line 10.

Also, it should be pointed out that Figs. 7 and 8, Page 150, are transposed.

—L. J. O'BRIEN
Gleason Works

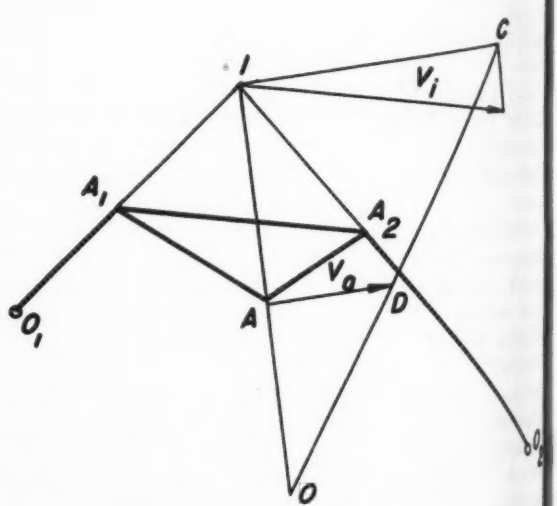
MACHINE DESIGN apologizes for the inadvertent transposition of the two cuts, and wishes to thank the author for pointing out the error.—ED.

". . . shortens acceleration analysi

To the Editor:

Recently I read Leon Beskin's article ""Simplified Me od Shortens Acceleration Analysis," in your June iss with great interest. Location of the center of curvat of certain points in a mechanism aids acceleration an ysis and provides the additional information necess when the ordinary step-by-step analysis fails.

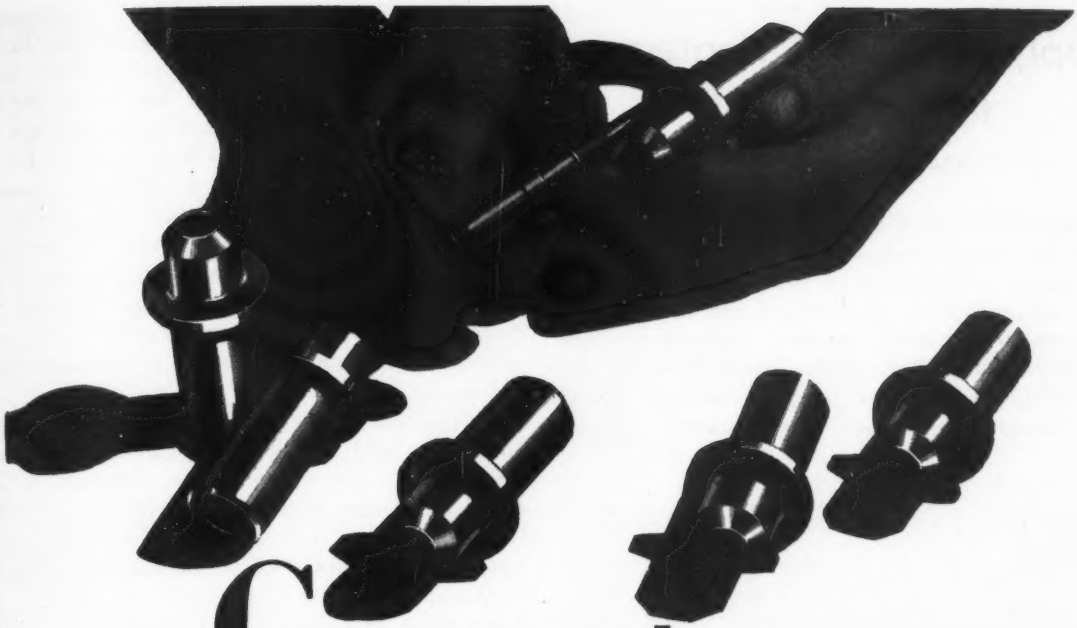
However, Mr. Beskin did not mention the purely gra ical method of locating the center of curvature. For ample, in the accompanying figure which is similar to t



problem illustrated in Mr. Beskin's *Fig.* 1, the center curvature *O* of point *A* is found at the intersection of li *IA* and *CD* when extended. The line *IC* represents component of the velocity (V_i) of the instantaneous cen *I* perpendicular to *IA*, and the line *AD* represents the ve ity (V_a) of *A*. The construction necessary to determ V_i and V_a are omitted for clarity.

The principle behind this construction is used by author in developing his formulas, and since it is gra ical and the ultimate acceleration problem is sol graphically, I think it deserves mention. The use of method in preference to the other would depend, I lieve, on one's preference for graphical or computati work.

—EDGAR SCHOU
Union Special Machine



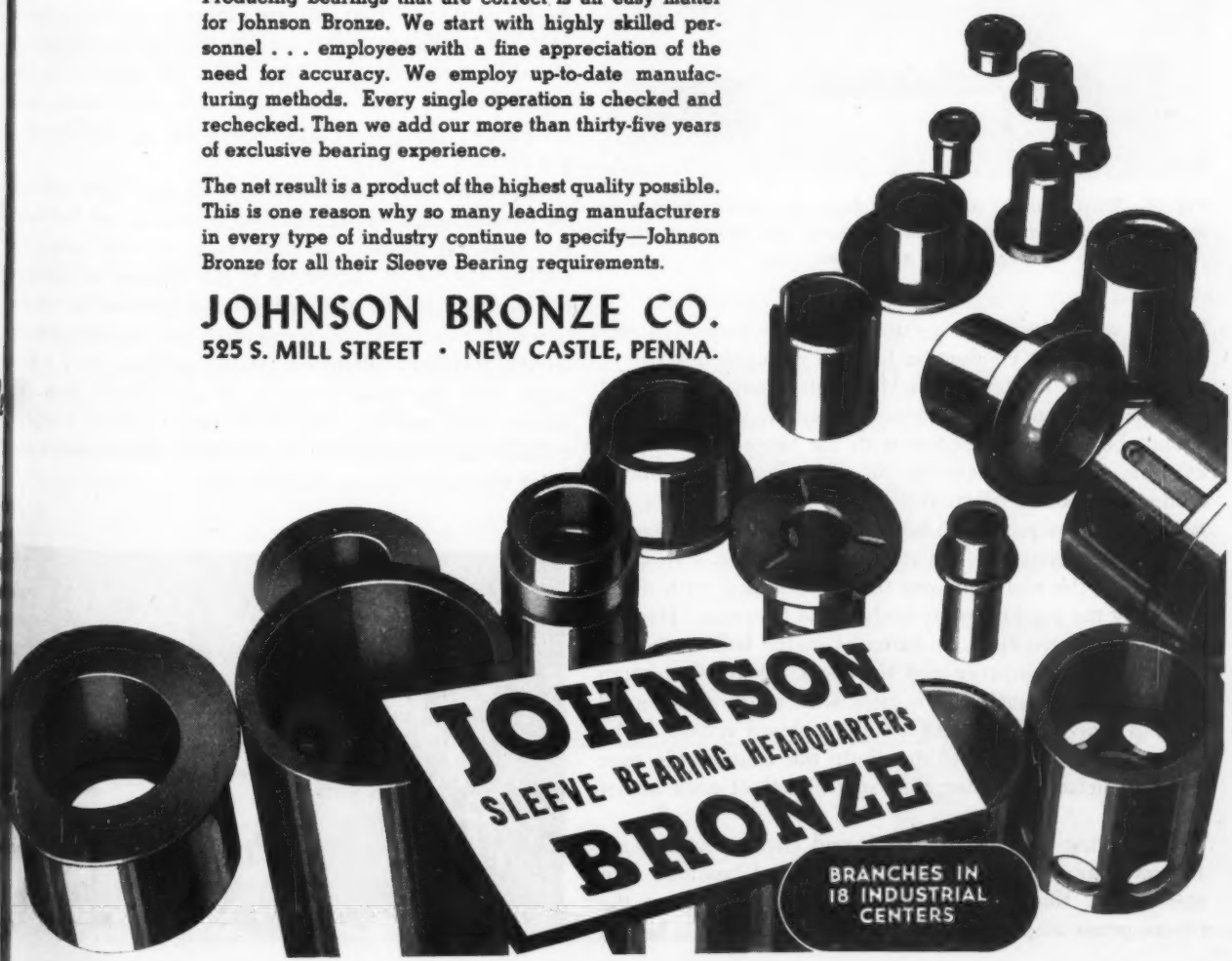
Correct

Sleeve Bearings cannot be "nearly right." In order to function properly . . . to deliver the utmost in performance for the longest period of time . . . they must be *correct* in every single detail. Every measurement must be precise. The alloy must be strictly according to specifications.

Producing bearings that are correct is an easy matter for Johnson Bronze. We start with highly skilled personnel . . . employees with a fine appreciation of the need for accuracy. We employ up-to-date manufacturing methods. Every single operation is checked and rechecked. Then we add our more than thirty-five years of exclusive bearing experience.

The net result is a product of the highest quality possible. This is one reason why so many leading manufacturers in every type of industry continue to specify—Johnson Bronze for all their Sleeve Bearing requirements.

JOHNSON BRONZE CO.
525 S. MILL STREET · NEW CASTLE, PENNA.



Applying Electric Heaters to Machines

(*Concluded from Page* 140)

clamped anywhere throughout their length and are not subject to the hazard of a short circuit if a conductor comes in contact with the unit, as with open-type heaters. Typical is an application of heating a roll too small to apply heaters to the inside surface. Strip heaters are mounted on the shaft and the small amount of heat is adequately furnished by radiation without the need of slip rings for contacts or other special arrangements. For roll

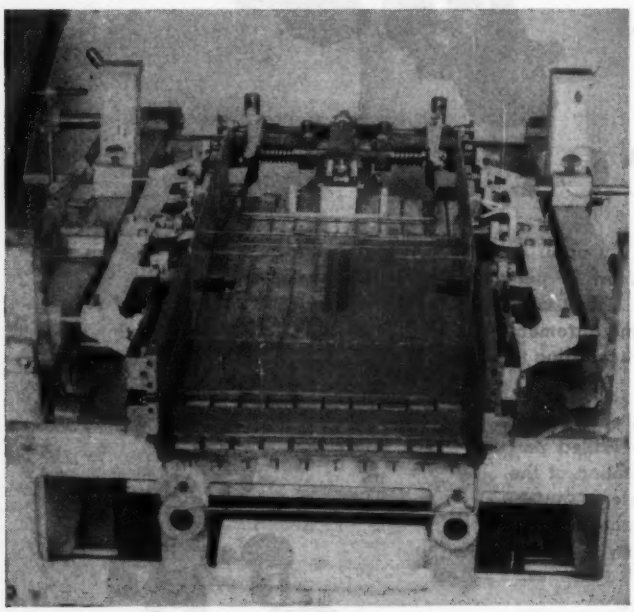


Fig. 8—Strip heaters on side folding and sealer plates as well as on bottom sealer plates melt the wax on bread wrapper to effect sealing

heating where low heat is satisfactory this method is recommended when conduction heating is impractical.

Convection method of heat transfer may be utilized when a self-contained system is desired, where heaters cannot be placed in contact with the heated part as, for example, in a small-diameter roll, and where the radiation method previously mentioned proves too inefficient or the temperatures required are beyond the economical capacity of a radiant system. The roll consists of two concentric tubes with the ends between the tubes sealed with disks, obviating the need of rotary seals at the bearings. Heaters of the immersion type are fastened in the hollow shell in any convenient manner and the leads taken off through slip rings. The enclosure is filled with a heat-transfer oil such as Dowtherm to form a fully enclosed system. The heat-transfer oil tends to distribute the heat evenly over the roll surface, reducing the "spot effect" of each heater element.

Circulating systems require forced circulation and are the most involved of the four methods of heating. For this reason, this system should be used only where the others prove inadequate or impractical. The oil is heated nearby with immersion heaters. Heat may be appl at a central reservoir or directly in the line. The la method has advantages in that less heat is lost throu radiation because the oil is not heated in large volu With this method, however, it is essential that the veloc of the oil through the heater is sufficient to prevent c bonization. To protect against this, baffles may be utili to increase the speed of oil flow over the surface of t heaters. Because losses increase as the system becom larger, it is highly desirable to design the system as co pactly as possible. The pump, its drive, fittings, seals a tubing required add to the costs and complication of t system.

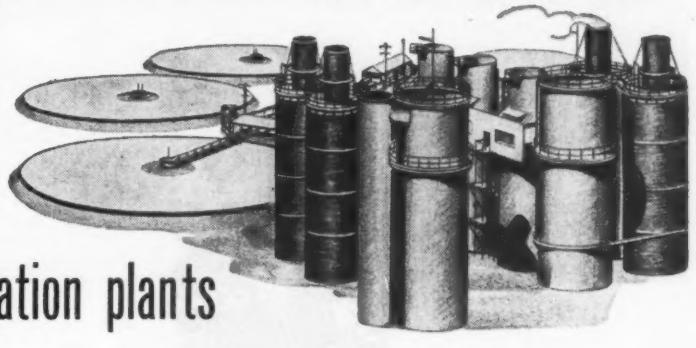
Cooperation of the following companies in supply information and illustrations is gratefully acknowledge American Machine & Foundry Co., Brooklyn, N. Y. (Fig 4 and 8); Anderson & Bolds, Cleveland; Cutler-Hamm Inc., Milwaukee; General Electric Co., Schenectady, N. (*Fig.* 2); The Hilliard Corp., Elmira, N. Y. (*Fig.* 7); H A Manufacturing Co., Buffalo, N. Y. (*Fig.* 6); Inter tional Cigar Machinery Co., Flemington, N. J. (Fig. Rockbestos Products Corp., New Haven, Conn.; Westi house Electric & Mfg. Co., Mansfield, O.; Edwin L. W gand Co., Pittsburgh (*Figs.* 3 and 5).

Most Maneuverable Weapon

FAST, light tank with automatic drive, eight spe forward and four reverse, mounts a 75-mm can and has the fire power equal to that of a medium ta Capable of speeds over forty miles an hour, it has be called by the Army "the fastest and most maneuverab combat weapon of its type in use by any belligerent o any front."

Replacing the M-5, the new M-24 has a low silhouet (its overall height being less than 87 inches and hull heig less than 60 inches), all important armored surfaces i cluding the turret sloped up to 45 degrees for protecti advantages, greatly reduced ground pressure in conju tion with new track and suspension, and doubled obsta climbing ability. Reduced ground pressure is of a lo value than the tread pressure of an average man. T power train utilized consists of two Cadillac V-type cylinder engines and two Hydra-Matic transmissions op ating through a power transfer unit.

When you turn to magnesium —turn to the pioneer producer and leading fabricator



Dow engineers and Dow fabrication plants are backed by 29 years' unmatched experience

When you work with magnesium, take full advantage of Dow's knowledge and experience in handling this lightest of structural metals. It pays, as many manufacturers well know. For Dow is the pioneer, with a background of 29 years' intensive work in producing, alloying and fabricating magnesium.

These years of development and production experience enable Dow to offer an exceptionally diversified magnesium line. But of equal or even greater importance to the manufacturer who buys this output is the wealth of technical, engineering knowledge which Dow likewise offers.

This knowledge is yours to command. When you build a product incorporating the benefits of weight-saving magnesium, feel free to call on Dow's trained staff of magnesium consultants for advice and assistance. Simply contact the nearest Dow office.

DOWMETAL

magnesium

THE METAL OF MOTION

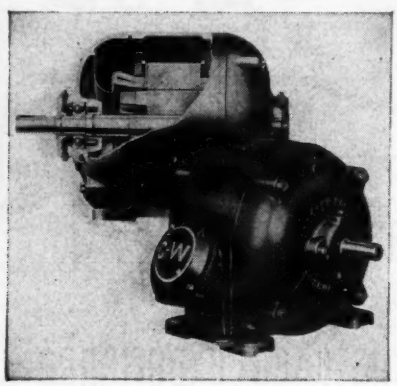
MAGNESIUM DIVISION, THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN

New York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

New PARTS AND MATERIALS

Protected-Type Motor

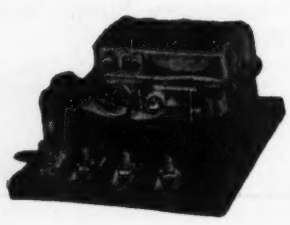
DESIGNED FOR both drip-proof and open-motor applications the new protected-type motor recently announced by the Crocker-Wheeler Division of Joshua Hendy Iron Works, Ampere, N. J., is constructed to provide protection against many of the most common hazards encountered in operation. The motors are rated on the same basis as conventional open types—40 degree Cent. rise, full-load continuous duty, with a 15 per cent service factor. Some of the features of this polyphase, squirrel-cage motor are the centrifugal bearing seals, aluminum diecast rotor, vinyl acetal protected windings, and the in-



ternally-shielded, partially-enclosed frames and shields which prevent metal chips, dripping liquid and other foreign matter from entering the motor. Because of the shielded-ventilation construction, the motors may also be used for many services for which other types of enclosures have been specified. At present the motors are available in sizes up to and including the 284 frame, with mounting dimensions conforming to NEMA standards.

Sensitive, Snap-Action Relays

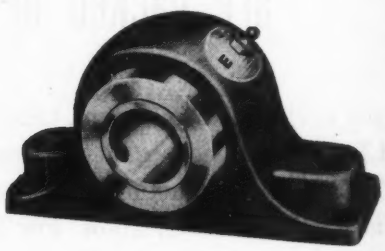
BECAUSE OF demand for sensitive, snap-action relays for a variety of applications where low costs and ease of adjustment are required, Struthers-Dunn Inc., 1321 Arch street, Philadelphia 7, has redesigned its Type 79XAX relay. Parts are accessible and sensitivity adjustments may be made easily in the new relay. Snap-action design assures full normally-closed and open contact pressures. The feature of the armature almost completing its travel in either direction before contacts snap into the new



position permits a broad range of use including vacuum tube circuits, overcurrent protection, pulsing circuits, and applications where close differential or extreme sensitivity is required. Contact ratings up to 10 amperes, 115 volt alternating current may be obtained. Sensitivity of .005 watt is obtainable from special units employing reduced contact pressure and rating.

Pillow Block Available

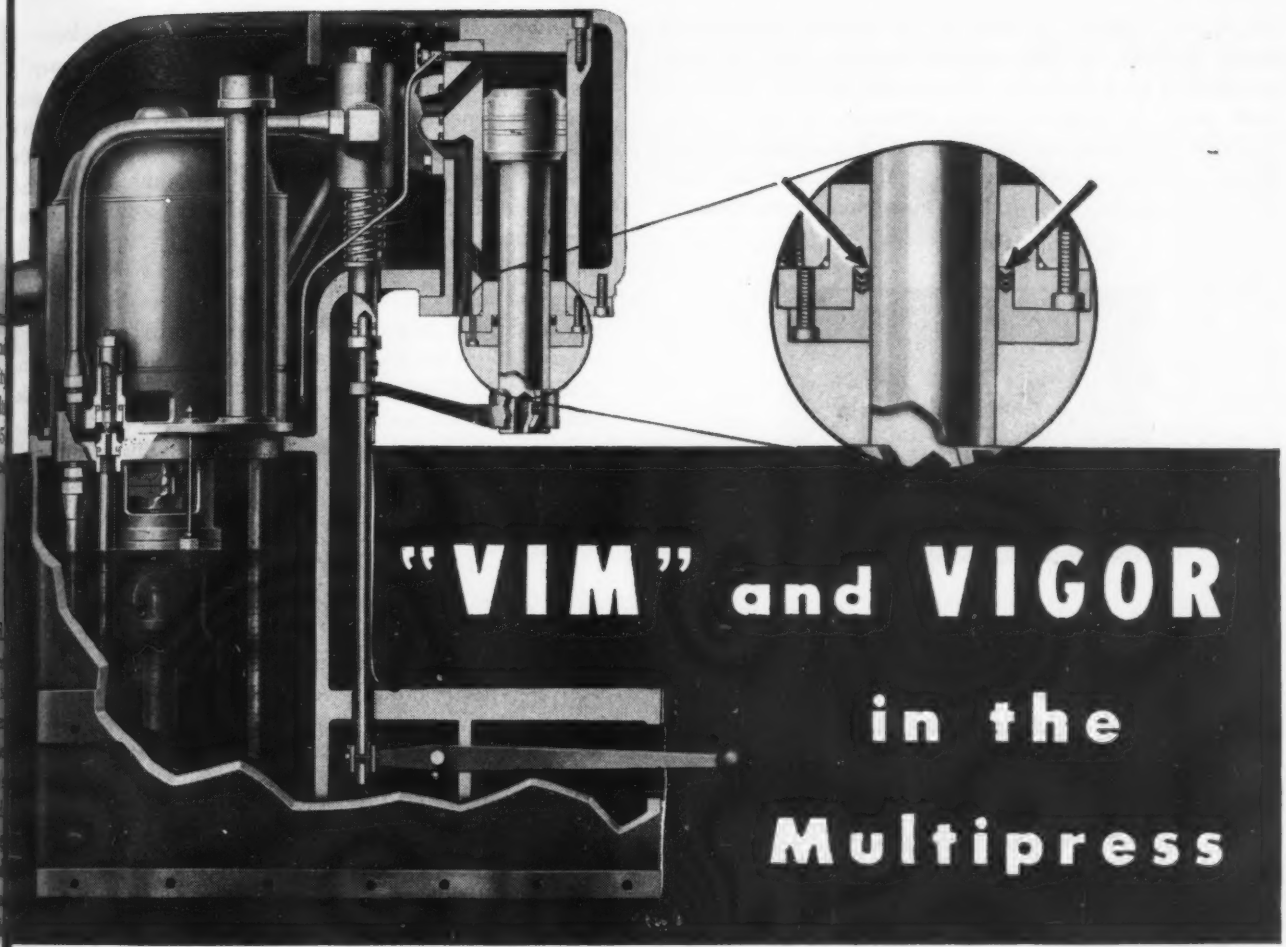
OFFERED BY The Dodge Mfg. Corp., Mishawaka, Ind., a new bearing known as Type "E" is now being furnished in both pillow blocks and flanged units. The unit is designed to give 30,000 hours or more of service under conditions for which it is adapted as determined by the use of standard selection tables applying to the company's line. Method of fastening to shaft is simple. On each end of the extended inner race is a steel collar having two headless set screws spaced 120 degrees apart and extending through clearance holes in the inner race to the shaft. These four set screws provide a positive lock to prevent rotation of the inner race on the shaft. The compact housings are precision machined to close tolerances. Labyrinth seals prevent loss of lubricant and



protect bearings against damage from dust, dirt or moisture. The bearings are assembled, lubricated and adjusted at the factory and while not fully self-aligning they have enough self-alignment inherent in the design and arrangement of the rollers to compensate for any normal misalignment.

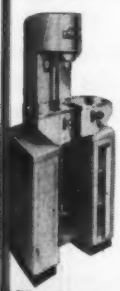
Quick-Fastening Lock Nut

FURNISHED IN two types—for sheet metal screws and for coarse machine screw threads—the new spring steel quick-fastening lock nut called Stalock announced by A-S Precision Products Corp., Burbank, Calif., offers 360 degrees contact with screw threads. This all-around contact allows positive gripping action and at the same time conserves tension locking power, thus permitting re-use of the nut. When screw is tightened, spring action



"VIM" and VIGOR in the Multipress

Why Denison selected VIM RESIN-IMPREGNATED "V" PACKINGS



This highly versatile machine tool typifies the progress made in hydraulic operation. Here is a compact press having endless uses, offering downward ram pressures of from 300 to 8000 PSI, upward pressures to 5000 PSI, fingertip control, automatic cycling, yet simple, lasting construction.

It is self-evident that the hydraulic oil used to provide speeds of from 20 to 200 inches a minute must be held in strict control by proper packing of the piston rod. After careful investigation of various types of packings, VIM Leather "V"s of the new resin-impregnated series originally developed for aircraft hydraulics were chosen. They hold a perfect seal at all working pressures from zero to 16,000 pounds, and more, and they're moulded so that the smooth grain side contacts operating surfaces, reducing friction.

Remember them, please, when designing new hydraulic equipment, and let us help you design a perfect packing installation.

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nut causes contact surfaces to be forced deeper into thread channels providing greater locking action and preventing nut from loosening even under extreme vibration. Tests show the fastener meets demands of light assemblies with performance exceeding AAF Specification No. 25533. For sheet metal screws, standard sizes are Nos. 4, 6, 8, 10 and 14; while for coarse machine screw threads,

sizes are Nos. 6, 8, 10, ¼ and 5/16-inch. Various shapes and sizes of special design can also be furnished on request. Recommended installations in aircraft include line supports, sealing-strip attachments, mounting of cowling panels, housing assemblies, attaching of nameplates, etc.

Two-Pole Circuit Breaker

SMALL IN size and light in weight, a compact, two-pole breaker has been added to the line of circuit breakers built by Heinemann Circuit Breaker Co., 113 Plum street, Trenton, N. J. Having an efficient blow-out action and a fast latch mechanism, the electromagnetic breaker illustrated is designed for service on 230-volt alternating current or 250-volt direct current of 50 amperes maximum. It may be connected either front or rear. Designed to carry full load continuously, it has an instantaneous trip at eight to ten times full load. The breaker also has a selection of three time delays, any one of which may be specified. Overall dimensions are: 5 1/4 x 2 11/16 x 2 inches.

Precision Remote Controls

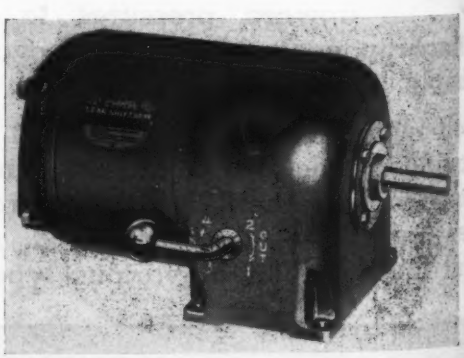
FOR WARTIME uses a new line of electric positioning has been developed under the Yardeny System by American Type Founders Inc., Remote Control Division, 11 West Forty-second street, New York 18. There are four basic types of these remote controls, which in tests have indicated their reliability under extreme conditions of temperature, humidity, vibration and shock. These are: A continuously variable control which is a nonsynchronous follow-up control provided with a repeat back indicator for positioning the shaft of a reversible motor to any point of any revolution. The second is a multiturn selector—a motor-driven device whereby its connected load may be placed in any one of several (usually six) adjustable positions over one or a number of revolutions. The third is a dual control which incorporates the features of both the continuously variable control and the multiturn selector. By this combination, the operator may position the load to any one of the preset positions, or by turning a knob to any point over the full range. The fourth is an integrating selector which provides automatic setting of the load to a large number of positions by means of a comparatively small number of pushbuttons. All the foregoing controls are light in weight, small in size and depend on electromechanical principles. They are built for any current characteristics, and their applications include the industrial, marine, aircraft and electronic fields.

Cored and Solid Bar Line Expanded

ADDITION OF several dozen new sizes of cored and solid bar stock in Oilite materials has been announced by Chrysler Corporation's Amplex Division, 6501 Harper avenue, Detroit 11. The new sizes embrace a range of ⅜-inch to more than 12 inches in diameter. Increased use of the self-lubricating Oilite bronze and ferrous base Super-Oilite in bar form were given as the reasons for the expanded size list. Most important advantages of the bar stock are the features of self-lubrication and high-load carrying capacities.

Selective Speed Gearshift

TO POWER MACHINE tools and other equipment requiring motor capacities of 1 horsepower or less, the Lima Electric Motor Co., 1701 Findlay road, Lima, O., has introduced its selective speed gearshift drive. This unit is rated at ½ and ¾ horsepower at 1200 revolutions per minute and at 1 horsepower at 1800 revolutions per minute, and has four speeds. Weighing only 85 pounds, the unit has an overall length of 18 inches, including the shaft extension; the overall width, including terminal box and shifting lever is 12 inches; while the overall height



is 9 inches. Identified as Type R, it is designed for continuous operation, with ample overload capacity. The company's semienclosed, drip-proof, ball bearing motor of either 2 or 3 phase, 50 or 60 cycle, 220, 440 or 55

Relays IN THE KEELER POLYGRAPH LIE DETECTOR

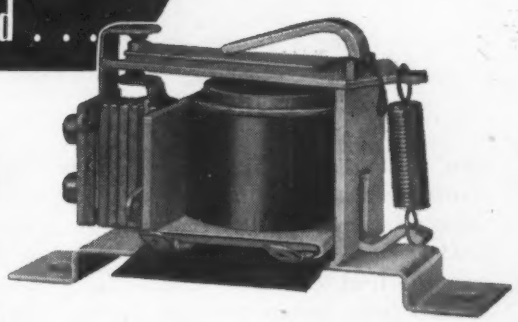
When faced with certain situations or conditions affecting the emotions, the body undergoes minute changes. The Keeler Polygraph (Lie Detector) detects, measures, and evaluates such changes. Recorded on a graph these changes provide clues which help solve baffling criminal cases. The Keeler Polygraph is built by Associated Research, Inc. of Chicago.

. . . wherever a tube is used . . .

THERE'S A JOB FOR *Relays* BY GUARDIAN

The Series 120 A-C Relay by Guardian is used in the Keeler Lie Detector as a control device for the amplifier in the electro-dermal recording section of the machine. This is a small, sensitive relay operating on any standard voltage up to 230 volts A-C. on any frequency between 20 and 150 cycles. Average power requirement of the Series 120 is 2 VA although where low current drain is an important factor, this may be reduced to a minimum of 0.2 VA.

This relay is especially designed for sensitive control of small circuit combinations where only a minimum amount of space is available. It is available for D-C operation in Series 125. Write for Bulletin No. 120 for A-C, or Bulletin 125 for D-C.



Series 120 A. C. Relay

Consult Guardian whenever a tube is used—however—Relays by Guardian are NOT limited to tube applications but are used wherever automatic control is desired for making, breaking, or changing the characteristics of electrical circuits.

GUARDIAN ELECTRIC

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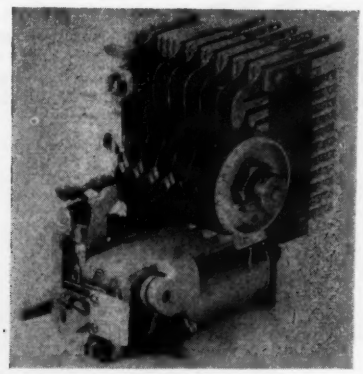
volts is furnished as a standard integral part of this unit. The drive is reversible in all speeds with the use of a reversing drum control. Single-phase units in above sizes, it is expected, will also be available when conditions permit. With evenly distributed speed changes, the ratios of Type R are 1 to 1, 1.33 to 1, 2 to 1, and 4 to 1. Shifting is done by a small lever which can be extended to any required position.

Selector for Automatic Control

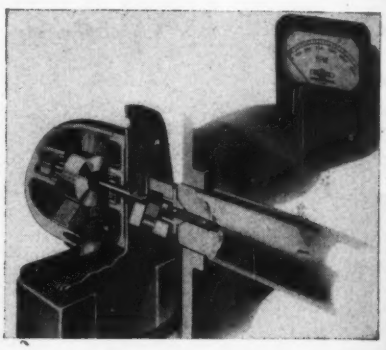
ADAPTABLE FOR use as an automatic or remote control device for railroad, radio, airport and many industrial applications, the compact FTR 800 automatic selector now being manufactured by Federal Telephone & Radio Corp., Newark, N. J., is offered for an operating life of 4,000,000 revolutions at speeds up to 60 steps per second. The unit will make connection between a number of given circuits and a similar number of other circuits, each preselected from a group; will control and perform various operations among a group of circuits by making consecutive individual connections; and will act as a timing device or switch when it is used in connection with time-pulsing apparatus. Also, it may be used in telegraph or electronic equipment, remote control and signaling sys-



tems, testing and radio control. Rotor assembly is operated by a stepping mechanism which responds to impulses of current. After each impulse a reed-suspended pawl engages a ratchet, moving wipers one step forward over the bank contacts. Connections are made by this method from circuits to other circuits. Stepping mechanism may be controlled manually by a dial or other means. Automatic control for the stepping magnet may be provided by interrupter springs, electronic circuits or relay circuits. According to the type of wiper used (double or single-ended), the selector can be arranged to have a capacity of from one to three 2-row levels of 22 points each or from one to six single-row levels of 11 points each. Dimensions of the selector are 2½ x 3¼ x 3½ inches.

Electric Speed Indicator

DEVELOPED BY Reliance Electric & Engrg. Co., 1074 Ivanhoe road, Cleveland 10, the new electric speed indicator gives accurate readings of speeds from 100 to 5000 revolutions per minute. The indicator consists of two units, the pick-up and the indicator. The pick-up unit, a miniature six-pole alternator with a permanent magnet rotor, is mounted on a shaft the speed of which is to be measured. The indicator, a permanent magnet moving-coil type, meters the pick-up output on a 3¾-inch scale which covers 95 degrees of arc. Resistance of the indicator has been made sufficiently high, so that the size or length of the leads connecting the units will have no effect on accuracy, and the indicator may be located at any distance from the pick-up unit. Instantaneous and continuous speed indication is independent of the direction of rotation of the shaft. Pick-up windings



are stationary. Foolproof shaft coupling is provided, and the indicator may be bracket or panel mounted. The indicators are provided for full-scale deflection corresponding to 1500, 2000, 2500, 3000 and 5000 revolutions per minute, and special scales are available to read other units.

Heavy-Duty Gathering Pump

INCORPORATING a number of changes and improvements, the new gathering pump announced by Blackmer Pump Co., Grand Rapids, Mich., has been designed for gathering in the oil fields. Its capacity is 90 gallons per minute; pressure, to 500 pounds per square inch; mechanical efficiency at 500 pounds per square inch, 85 per cent and suction pressure, 0 to 100 pounds per square inch. The pump is powered by an engine or motor. Designed especially for outdoor service, the pump is of particularly rugged construction.

Synthetic Rubber Adhesives

SYNTHETIC RUBBER adhesives have been announced by The B. F. Goodrich Co., Akron, O., which have already been used on life-saving boats, tank crash pads, pontoons, tank linings, brake tubes, etc. One of the types is for general utility use and will adhere to almost any clean surface. Synthetic cements are also available for heat vulcanization, air curing or cold adhesions. Compounded synthetic cements are suitable for fabric, leather and synthetic adhesions, to themselves or to each other. These adhesives have a nontoxic solvent, and will form a bond with a large variety of materials. For metal adhesion, two special cements have been developed, known as the Plastilock 300 Series. The first is used with

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A business institution acquires a fine reputation, not because of its size, not because of the dividends it pays, not because of the patents it holds; but because of the acceptance of its products and methods of doing business by prospects, by customers, and by its employes.

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This is evidenced by the Navy's acceptance of Falk products since before World War I; by the attitude of the Falk management toward Falk employes during the lean years of the economic cycle; by the honors bestowed upon Falk officials; by recognition of the high standards of its employe relations accorded by a national magazine; and by interplant and office activities of the employes themselves.

It has long been an axiom of the Falk management that "to build a better product, first improve the man." Falk quality is due in no small measure to the thoroughness of Falk apprenticeship courses.

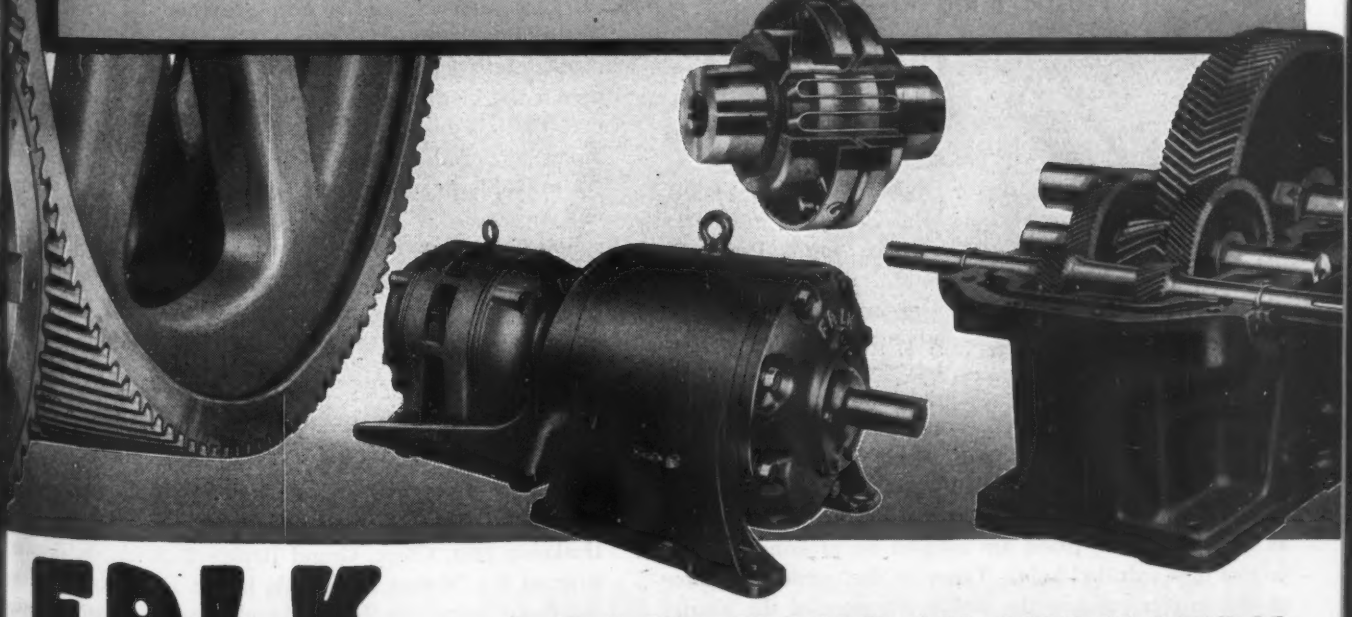
The same thoroughness employed in molding men is employed in molding Falk material. Falk casts its own steel in its own foundry to its own specifications.

To secure the watch-like precision so characteristic of all that Falk produces, it has always been the Falk policy to design its own machinery.

Because Falk has always known its men, its material, and its tools, it logically follows that Falk engineering has always been better than it could have been, had this engineering been done with unknown components.

All of this is just plain good business. No reputation can ever be acquired, much less maintained, unless it is based on plain good business. For plain good business is business that is good first for the customer, then for the employe, then for the corporation—and only then for the management.

This is why Falk is "a good name in industry".



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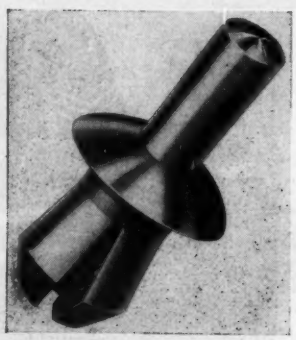
vulcanized neoprene when it is bonded to porcelain, metal, etc., while the second is used with uncured neoprene for the same purpose.

Engine Hour Meter

AN ELECTRICALLY operated engine hour meter for indicating hours, and accumulated number of hours, of operation for trucks, tractors, marine motors, any unit driven by diesel or gasoline engines, and for use by laboratories or on testing work, has recently been announced by John W. Hobbs Corp., Springfield, Ill. Hours of operation are indicated by three hands. One of these hands turns one complete revolution in 100 hours, the

second, one revolution in 1000 hours, and the third, one revolution in 10,000 hours. The engine hour meter, mounted on the instrument panel, operates through the ignition switch or by use of a special pressure switch. It is available in direct current in ranges from 6 to 115 volts; in alternating current in 110 volts, 60 cycle and 220 volts, 60 cycle operation. Overall dimensions are 2⅞ x 2½ inches.

Plastic Blind Rivets



IDENTIFIED as "Des-Rivets", the new plastic blind rivets developed by the Plastic Development Division of the Victory Mfg. Co., 1105 Fair Oaks avenue, South Pasadena, Calif., are based on a wedging action and take advantage of flow characteristics of plastic materials under pressure. They are molded in one piece, consisting of a head with plug attached by a thin breakaway section and a tapered shank split to form four tapered fingers. Shank and head are hollow to the same diameter as the plug. The rivets are applied by pressing tapered fingers into a drilled hole. Taper on the outside diameter of the fingers reduces the inside diameter of the shank, the rivet and work being held in place by pressure of the depressed fingers. Impact from rivet gun shears the plug and drives it into the plastic shank until the plug is flush with both ends of the rivet, maintaining the contour of the rivet head. Wedge action of the plug in the tapered shank expands the fingers against the wall of the drilled hole and upsets the shank end of the rivet. The fact that the fastening creates its own upset end by flow of the plastic allows for a wide variation in the thickness of the materials being used. Rivets may be single inserted or assembled in "sticks" by inserting the undriven plug of one rivet into the shank of another. An automatic rivet gun is available to accommodate sticks of rivets. A wide variety of shapes and sizes can be furnished in several plastic materials including Nylon, and in all conventional and many special colors. Physical properties, of course, depend upon the plastic from which the rivets are molded.

Heating Units Announced

TO MEET THE more specialized requirements of extended surface heating coils in air conditioning, heating, drying and processing systems, the Modine Mfg Co., Racine, Wis., has developed a new and broader line of copper blast heaters and booster units. Available now for wartime priority applications, these heating coils incorporate design modifications which make them more adaptable to the actual needs of heating engineers and drying equipment designers. The new coil design features all-copper and copper alloy condensers, die-formed fins permanently bonded to tubes by metal, pressure-resisting brazed construction of steam carrying passages, and a new provision for even steam distribution within the coils. Casings are designed for duct installation in vertical or horizontal positions and permit use of ducts having cross-sectional areas no larger than the face of the coils. New baffle arrangement helps direct flow of air through the coil. Standard blast heaters are available in seven types (combinations of tube depths and fin spacings), 17 lengths and 3 widths. Steam distribution blast heaters are furnished in 4 types, 9 lengths and 3 widths. Booster units can be supplied in 3 types, 8 lengths and 4 widths. This provides 561 individual coils from which to select in meeting exacting requirements.

Super-Sized Plywood Panels

SUPER-SIZED plywood panels are being produced by Haskelite Mfg. Corp., Grand Rapids 2, Mich., in sizes as large as 8 x 50 feet and in 1¼ inches in thickness. The one-piece large panels are bonded with phenolic resin glues in a single cycle, producing a uniform and stabilized panel of high structural strength. They are available in

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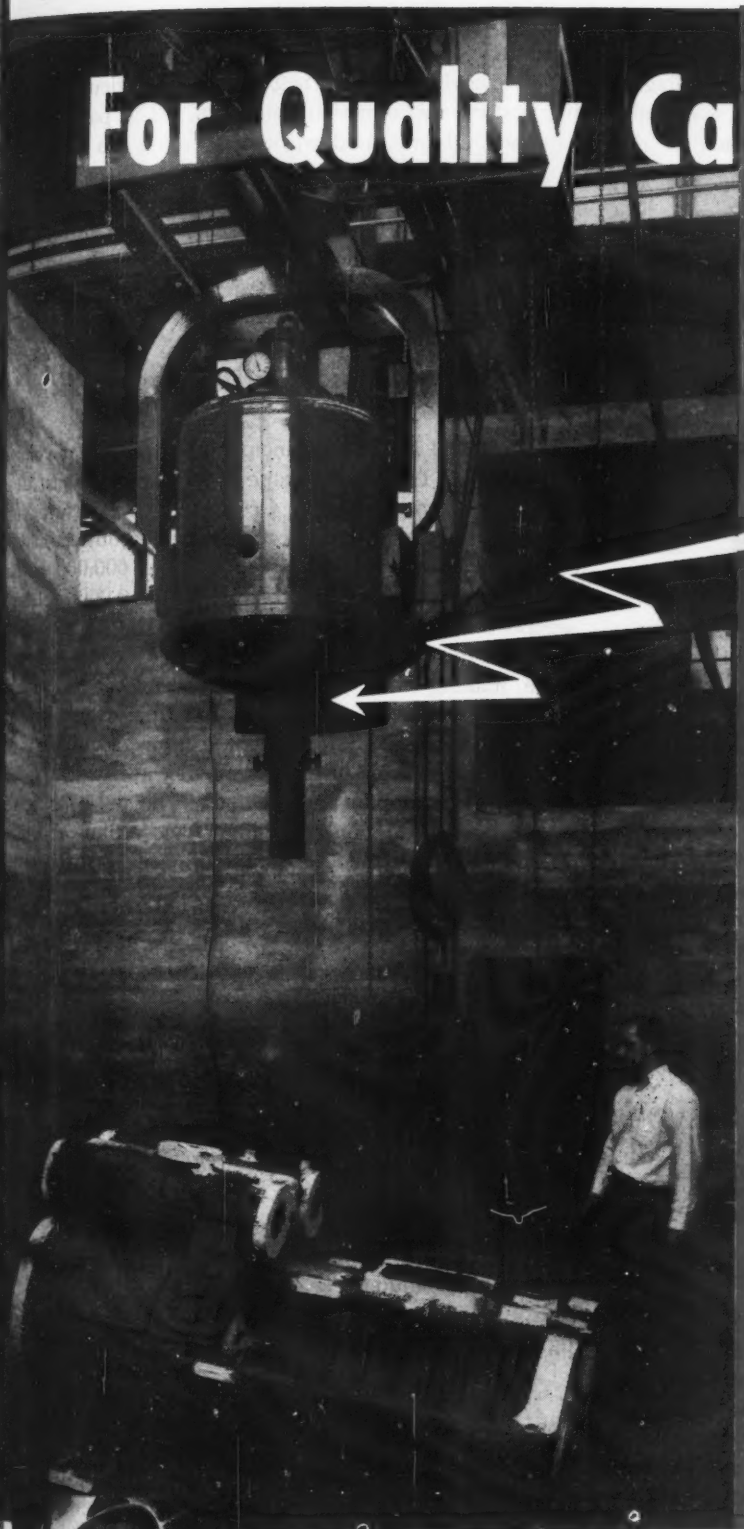

In a special section of PSF's laboratories, within thick concrete walls, one of the latest GE million-volt X-ray machines furnishes the final proof of the uniform soundness of PSF castings. It's a searching eye from which no defect is hidden, however deeply located. Used both as a special assurance on individual jobs, and as a running check on all production work, this modern "inspector" helps to hold *all* PSF output to consistently high standards. No matter what *your* casting requirements are, PSF's modern facilities can give you visible proof of quality—*in advance!*

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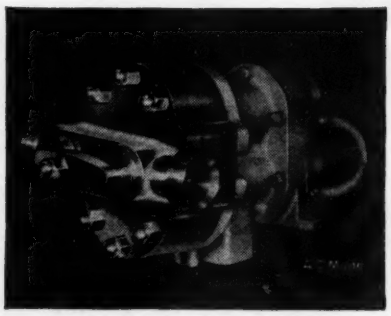
plywood construction (Plymold) or plastic-coated (Plastiply). For boat planking and decks, bus, truck, trailer and train floors and sides, this panel offers many advantages. Structural characteristics eliminate the need of many ribs and other structural members in the fabrication of the panels.

High-Vacuum All-Purpose Pump

KNOWN AS THE Flow-Master, the new all-purpose pump of Marco Co. Inc., 511 Monroe street, Wilmington 17, Del., will transfer, meter or proportion any material, light or heavy and viscous. So constructed that it can be disassembled, cleaned and reassembled quickly, this



positive displacement pump maintains a steady, nonpulsating delivery. Rotors in the pump are hydraulically balanced; thus friction at this point is negligible. Seals are of single-gland type, housed in stainless steel armor. Standard pumps are built in capacities for all standard ranges, and are of stainless construction unless otherwise specified.

Engineering Dept. Equipment

3-Way Artists' Board

COMBINED IN one unit by Engineering Manufacturing Co., Sheboygan, Wis., is a revolving drawing board on a base adjustable to the angle, height and working position required. The board swivels on a strong steel center pin, and special friction inserts assure position at any point of a 360-degree arc. A steel bracket gives positive adjustment and locking to the work angle required, and a spring-action clamp permits a shifting range of 8½ inches up or down of the working table on the base.

New Photographic Paper

RECENTLY PERFECTED by Peerless Photo Products Inc., New York, a new type of photographic paper will enable the blueprint operator, without the installation of a darkroom or special printing equipment, to make black and white photographic reproductions on either paper or transparentized vellum. Blueprints, old yellowed drawings or weak pencil tracings can be copied; also high-contrast, litho-type paper negatives can be made even from pencil drawings. Such negatives are particularly useful for positive reproductions on cloth. The paper can be handled and processed without dimming the usual lights in the room, and has new antidiffusion characteristics to improve definition and permit full, noncritical exposure without closing up or losing fine lines. It is also reported to be a good medium for copying letters or documents as they produce a fine reflex-type negative.

Stroboscope Developed

BASED ON A NEW principle, the recently developed CML Model 1200 stroboscope of Communications Measurements Laboratory, 120 Greenwich street, New York, increases the range through which moving objects may be examined. Rotary speeds from 600 to 600,000 revolutions per minute or vibrations from 10 to 10,000 cycles per second can be stopped and studied. The light source is mounted in a small probe at the end of a 5-foot flexible cable, making small objects readily visible at close range. Provision is made to operate the unit from external tuning fork or crystal standards where extreme accuracy is required. The motion of objects moving at irregular speeds may also be stopped and viewed. The pulse being derived from a stable audio oscillator, an accurate repetition pulse rate is obtained. A light intensity control switch provides for greater flexibility, enabling the user to control both the light intensity and the duration of the pulse length.

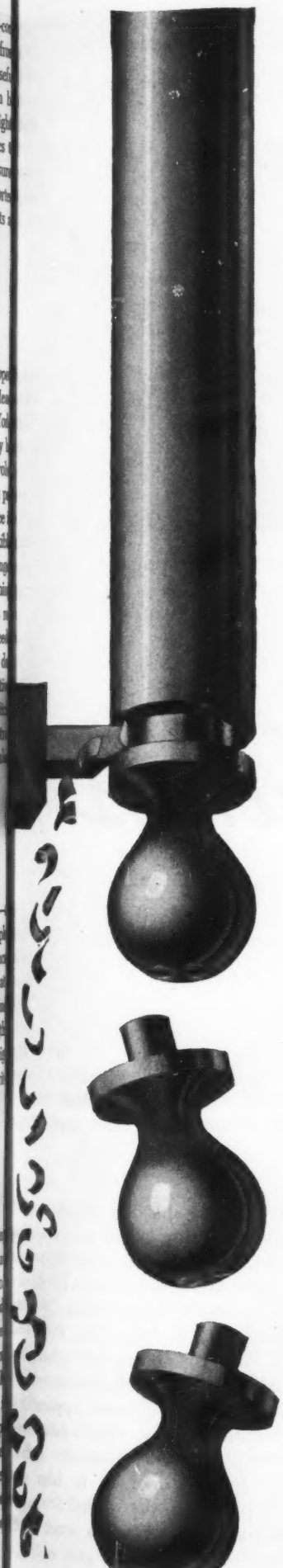
Multiple Carbon Pencils

MADE IN THREE degrees—medium, firm and hard—a new special type of pencil known as DuroLead Multiple Carbon Copy No. 700 has been developed by Reliance Pencil Corp., Mount Vernon, N. Y. It is made with a patented lead and exclusive process of bonding wood and lead to give the pencils 25 to 75 per cent greater strength. The pencil is smudgeless, making clear, easy-to-read originals and six or more carbon copies. Regardless of writing pressure, the lead is said not to cut the top sheet.

Tape Dispenser Offered

AN IMPROVED MODEL of Tapemaster, for dispensing paper, cellophane or cloth adhesive tape, has been announced by Bauer & Black, Division of The Kendall Co., 2500 South Dearborn street, Chicago 16. Among the advantages of the new model is its all-metal, sturdy construction. The cutting knife is attached to the rocker arm removable to facilitate loading. This arm provides a braking surface to grip the tape, while the desired length is flicked off. When the brake releases, a short tab of tape is ejected for the next use. A new suspension arrangement permits use of any sized tape or combination of sizes to a total tape width of 4 inches. Also, rolls of tape wound on cores of any diameter larger than 1 inch can be used.

J&L STEEL



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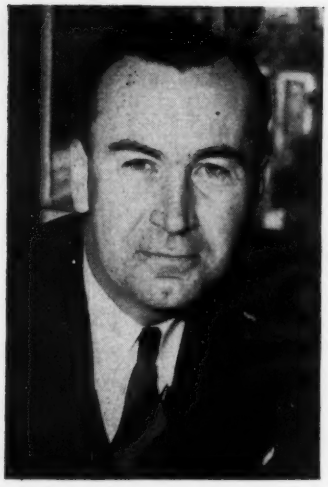
JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PENNSYLVANIA



Francis J. Rouan

S. L. Eastman

Clinton R. Hanna

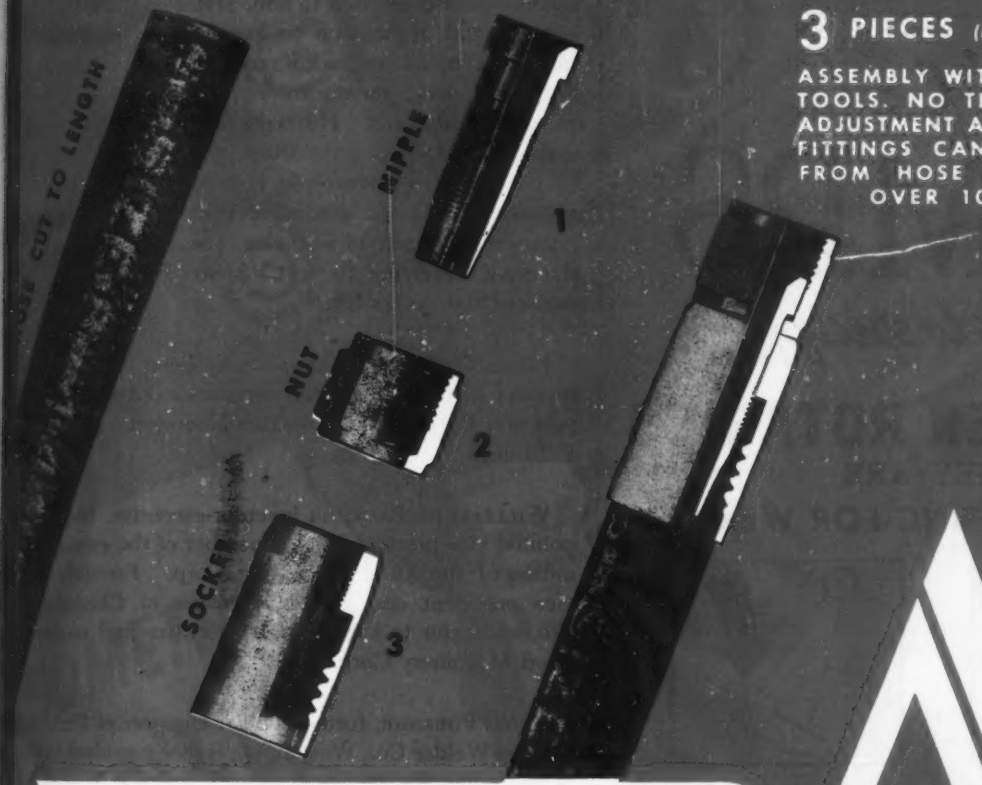
MEN...of...machines

FRANCIS J. ROUAN has reached the top creative engineering post at Pitney-Bowes Postage Meter Co. in being appointed chief development engineer for the company. Coming "up-from-the-ranks" Mr. Rouan began his career at the age of seventeen as a cub-draftsman in 1922, and in the ensuing twenty-three years he has tackled practically every job in the development engineering department which he now directs. Most of his work has been of an all-purpose character such as is typified by the activities of the drafting, experimental and development engineering departments of medium-sized specialty manufacturers. His many years of work on the company's various postage meter machines and special post office machines brought recognition in 1939 with the appointment as assistant to the chief designer, the late Linden Thatcher, whom Mr. Rouan now succeeds. Two years later—again under Mr. Thatcher—Mr. Rouan was appointed assistant chief engineer in charge of design, from which post he was advanced to chief development engineer.

S L. EASTMAN has recently been named chief engineer of The Cleveland Worm & Gear Co., Cleveland. His connection with the company dates back to his graduation from Case School of Applied Science in 1927. Since then he has served in various capacities, his first assignment being in the testing laboratory. Later he was made sales engineer and assistant sales manager. In April, [illegible] he was promoted to the position of [illegible] ant chief engineer, in which capa[illegible] he served until his present appointm[illegible]

C LINTON R. HANNA, inventor of [illegible] tank-gun stabilizer which enables A[illegible] tanks to fire accurately while in mo[illegible] has been appointed associate dire[illegible] of the Research Laboratories, West[illegible] house Electric & Mfg. Co., East P[illegible] burgh, Pa. Mr. Hanna who since 1[illegible] has been associated with the compan[illegible] the development of new apparatu[illegible] also manager of the electro-mecha[illegible] department of the Research Lab[illegible] tories. He has credited to him m[illegible] than 80 patents in the United States a[illegible] foreign countries. For his work in [illegible] development of the tank gun stabili[illegible]

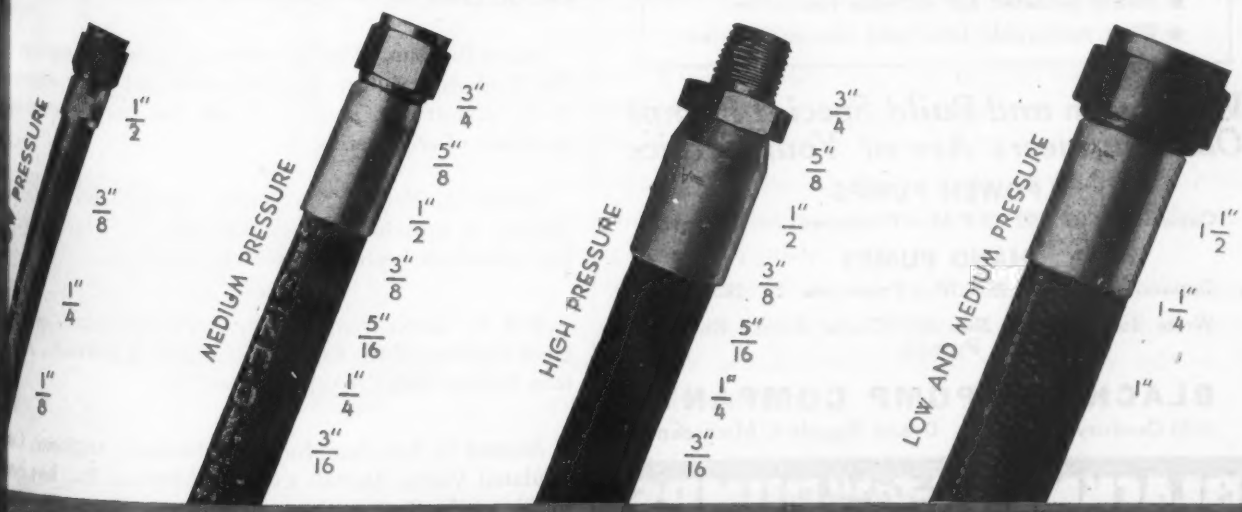
WAR TESTED ON AIRCRAFT ALL OVER THE WORLD
NOW AVAILABLE FOR INDUSTRIAL APPLICATIONS



3 PIECES *(EACH REPLACEABLE)*

ASSEMBLY WITHOUT SPECIAL TOOLS. NO TIGHTENING OR ADJUSTMENT AFTER ASSEMBLY. FITTINGS CAN BE REMOVED FROM HOSE AND RE-USED OVER 100 TIMES.

AEROQUIP CORPORATION
JACKSON, MICHIGAN, U.S.A.



★ ★ 303 WAREHAM BLDG., HAGERSTOWN, MD. - 1709 W. 8th., LOS ANGELES ★ ★ ★

WHEN COST-CUTTING MACHINES AND PROCESSING EQUIPMENT—

Require

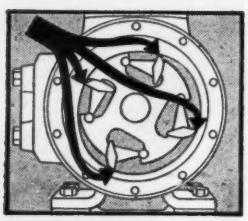
PUMPS

be sure to specify

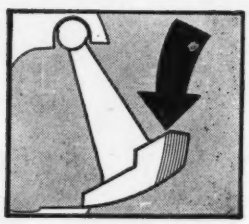
BLACKMER ROTARIES

. . . THEY ARE

SELF-ADJUSTING FOR WEAR



Wear is confined to these points . . . the tips of the "buckets."



This much can wear away without affecting the capacity of the pump.

"BUCKET DESIGN"

(swinging vane principle)

is a Blackmer cost cutting feature.

STANDARD CONSTRUCTION

- Unlined for oils, syrups, etc.
- With removable liners for corrosive or abrasive liquids
- Steam-jacketed for extreme viscosities
- With removable liner and steam-jacketed

We Design and Build Special Pumps Our Engineers Are at Your Service

POWER PUMPS

Capacities to 750 G.P.M.—Pressures to 500 psi.

HAND PUMPS

Capacities to 25 G.P.M.—Pressures to 125 psi.

Write for Bulletin No. 306—Facts About Rotary Pumps.

BLACKMER PUMP COMPANY

1970 Century Avenue Grand Rapids 9, Michigan

BLACKMER Rotary PUMPS

"BUCKET DESIGN"—SELF-ADJUSTING FOR WEAR

Mr. Hanna was awarded a Presidential Citation in 19[illegible]. His work also includes the design of an automatic volt[illegible] regulator which is applied to motors, turbines and [illegible] devices, including the tank-gun stabilizer. The direct[illegible] of development work on the Westinghouse Photophone[illegible] one of the first successful methods of producing sou[illegible] movies—is also credited to him. Prior to 1928, Mr. Ha[illegible] was engaged in research on loud speaker equipment a[illegible] on power tubes for radio receiving sets. Later he [illegible] vestigated noise measuring apparatus and methods [illegible] quieting equipment. His work with gyroscopic contr[illegible] basis of the tank-gun stabilizer, goes back to 1936 wh[illegible] he developed a gyroscopic regulator for steel mill [illegible] motors. Mr. Hanna graduated from Purdue university [illegible] 1922 with a degree of bachelor of science in electrical [illegible] gineering, receiving the professional degree of electr[illegible] engineer four years later.

♦

JAMES C. HARTLEY has recently been made director [illegible] research of the Heppenstall companies, Pittsburgh, Brid[illegible] port and Detroit. The research laboratories are located [illegible] Pittsburgh.

♦

WILLIAM DE BACK, an inventor-executive, has been [illegible] pointed vice president and a member of the executive co[illegible] mittee of the Food Machinery Corp. Formerly he w[illegible] vice president and general manager of Chisholm-Ry[illegible] Co., and prior to that was an executive and engineer [illegible] Food Machinery Corp.

♦

E. W. FORKNER, formerly chief engineer of Federal M[illegible] chine & Welder Co., Warren, O., is vice president in cha[illegible] of operations for the newly formed Precision Welder & M[illegible] chine Co., Warren, O.

♦

ZOLLY CARLTON VAN SCHWARTZ has left the post of [illegible] gineering coordinator, Firestone Aircraft Co., and of m[illegible] chanical research and development engineer for Firesto[illegible] Ordnance, to join Peck Stow & Wilcox Co. as chief of m[illegible] chinery and development in the technical division.

♦

CHARLES E. MYLES has become design and project e[illegible] gineer of Durasteel Co., Hannibal, Mo. Previously he h[illegible] been design and research engineer of McQuay Norris M[illegible] Co., St. Louis 10.

♦

JAMES B. FORD, formerly senior aeronautical engineer [illegible] the Civil Aeronautics Administration and chief engine[illegible] Doak Aircraft Co., Torrance, Calif., has been elected vi[illegible] president of engineering.

♦

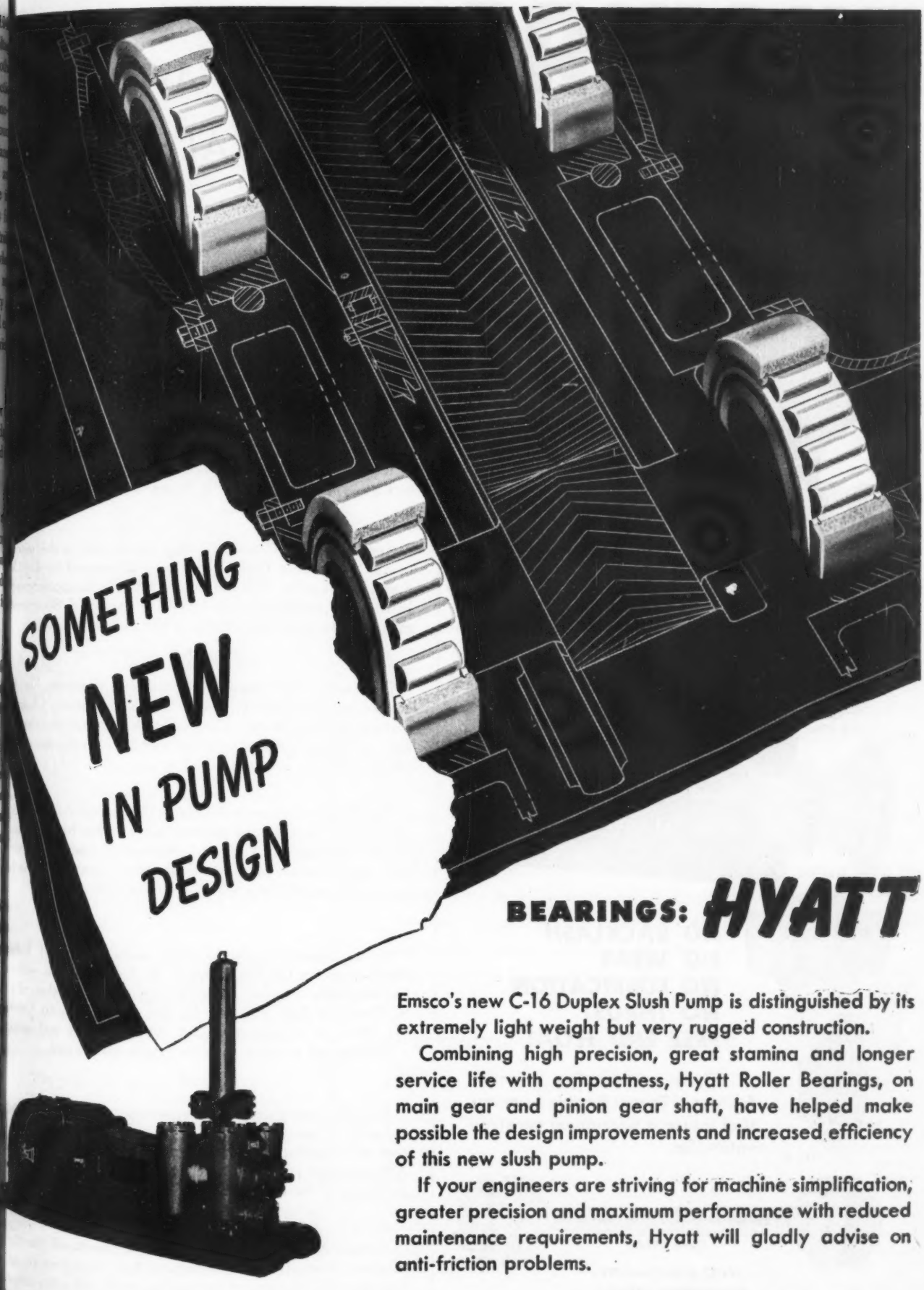
LEROY A. PETERSEN succeeds the late Jesse H. V[illegible] Alstyne as president of Otis Elevator Co. Mr. Peters[illegible] had previously been executive vice president.

♦

ROY P. TOOKE has recently been appointed assist[illegible] chief engineer of the General Engineering Division, Am[illegible] ican Rolling Mill Co., Middletown, O.

♦

ROBERT C. KELLOGG, formerly standards engineer, Co[illegible] solidated Vultee Aircraft Corp., Allentown, Pa., has be[illegible] made production engineer, Perfection Stove Co. In[illegible] Cleveland.



BEARINGS: **HYATT**

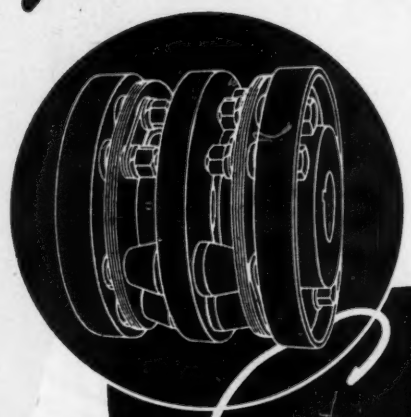
Emsco's new C-16 Duplex Slush Pump is distinguished by its extremely light weight but very rugged construction.

Combining high precision, great stamina and longer service life with compactness, Hyatt Roller Bearings, on main gear and pinion gear shaft, have helped make possible the design improvements and increased efficiency of this new slush pump.

If your engineers are striving for machine simplification, greater precision and maximum performance with reduced maintenance requirements, Hyatt will gladly advise on anti-friction problems.

HYATT BEARINGS DIVISION, GENERAL MOTORS CORPORATION
HARRISON, NEW JERSEY

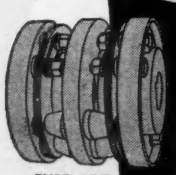
THOMAS
Flexible COUPLINGS



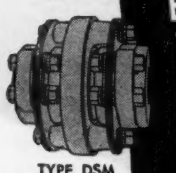
Eliminate:

BACKLASH, FRICTION, WEAR AND CROSS-PULL

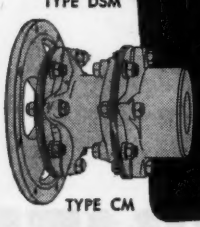
the four destructive evils found in other types and makes of couplings.



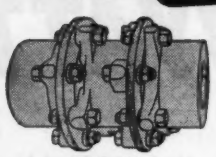
TYPE DBZ



TYPE DSM



TYPE CM

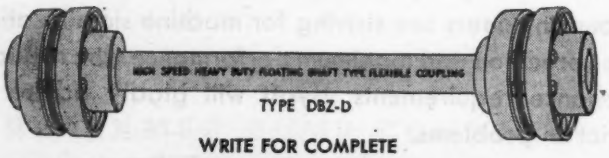


TYPE AM

NO BACKLASH
NO WEAR
NO LUBRICATION
NO THRUST
FREE END FLOAT

These are the five essential features of Thomas Flexible Couplings that insure a permanent carefree installation.

TYPE SS



TYPE DBZ-D

WRITE FOR COMPLETE ENGINEERING CATALOG

THOMAS FLEXIBLE COUPLING CO.
WARREN, PENNSYLVANIA

BUSINESS AND SALES BRIEF

CONNECTED with the company since 1934, Robert Bockrath has been made manager of magnesium s for the Houston, Texas, office of The Dow Chemical Co., land, Mich. Mr. Bockrath will operate under Arthur Smith southwestern sales manager of the magnesium division, has headquarters in St. Louis.

♦

Addition of two new engineers to the staff of the De office has been announced by Kennametal Inc., Latrobe, They are: Leonard K. Weeks, previously service engineer Cincinnati Milling Machine Co., and John R. Gough, form tool supervisor for George D. Roper Corp.

♦

Appointment of Charles C. Bray as assistant to the mana of the Worm Gear Division has been announced by De L Steam Turbine Co., Trenton, N. J. Prior to his appointme had been a metallurgist for Joseph T. Ryerson & Sons, ser in the special steel department of the Philadelphia office.

♦

Previously sales manager, Walter W. Bertram has b elected vice president in charge of sales for Morse Chain division of Borg-Warner Corp. Replacing him as sales m ager is Robert J. Howison, who had been in charge of m motive sales.

♦

Establishment of an Indianapolis district sales office in Su 831-3 of the Architects & Builders building has been announ by Inland Steel Co., Chicago. Norbert E. Smith, who ser as a representative of the Chicago office since 1936, has b appointed district sales manager of the new office.

♦

To its rapidly expanding field engineering staff Eutec Welding Alloys Co. has added C. F. Patterson who will s the Michigan territory. Another appointment is that of J A. Owen as field engineer for the state of North Caro Mr. Owen is a veteran welder, having owned and operat a welding and machine shop for twenty-eight years.

♦

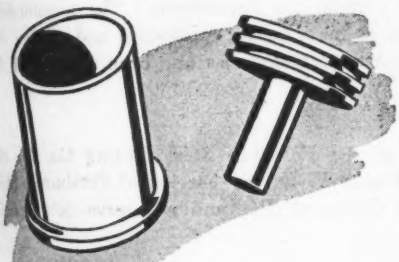
Recently announced is the appointment of H. C. Allin as assistant general sales manager of Wickwire Spencer and its subsidiary, the American Wire Fabrics Corp. Allington had served in the capacity of sales research engi in charge of the development and expansion of markets.

♦

Gerotor May Corp., Baltimore, has purchased the Co Machine Products Co., Logansport, Ind., manufacturer of and hydraulic equipment for machine tools and other indus applications. John C. Cotner, organizer of the Cotner C pany, was elected a vice president and director of the Ge May Corp. shortly after the transaction was completed. Ro Esser, formerly chief engineer of Logansport Machine Co.

ADD YEARS OF SERVICE TO WORKING PARTS

USE STAINLESS — SURFACE HARDENED



"YOU SOLVED OUR PROBLEM!"

says the chief engineer of a prominent manufacturer of pressure reducing valves

"Surface hardened stainless steel proved to be the answer for prolonging the life of the working parts of our pressure reducing valves —even though the following hazards always are present:

"Dry, sliding friction under high temperatures rasps away at the surface of cylinder lining and piston wall . . . high pressure steam at 600 psi. develops pressure-cutting velocities to ruin valve seats . . . heat—750° of it—places heavy strain on valve stems and close fitting surfaces.

"The glass-hard surface provided by the Stainless Surface Hardening Process plus the inherent corrosion resisting qualities of Stainless Steel licked our problem, I'm glad to say."

Combining the corrosion resistance of Stainless with the abrasion resistance of a nitrided surface, the Stainless Surface Hardening Process minimizes operating hazards likely to endanger vital working parts of your product.

This efficient new process provides stainless steel with a "glass-hard" (91 to 97 Rockwell 15-N) surface. Dimensional characteristics are not appreciably affected . . . for parts machined to almost exact dimensions need only a light grinding or lapping after processing.

Both austenitic and martensitic grades of stainless steel now can be surface hardened with this process. Highly stressed parts in which heat treatment has developed high tensile strength, ductility and impact resistance, can be surface hardened by this process to insure great abrasion resistance, without altering core properties.

Surface hardening will extend the service life of your stainless steel valves, pumps or instrument parts . . . write or call today for detailed information.

EVERYTHING IN STAINLESS STEELS!

Whatever your Stainless requirements—sheets, bars, tubing, wire, valves, fittings, even bolts and nuts—the inventory of Industrial Steels, Inc. is complete and comprehensive . . . and you'll receive same-day attention to your order. One way to save valuable time and needless bother . . . one way to speed all your Stainless requirements to your plant . . . is to place your entire order with Industrial—where the largest and most diversified warehouse stock of stainless steels in America is maintained. Expert metallurgists at Industrial gladly will assist you with any Stainless specification or fabrication problem. For complete catalog, write Industrial Steels, Inc., Cambridge 41, Mass.

SSCoH

STAINLESS SURFACE HARDENING CO.

255 BENT STREET, CAMBRIDGE 41, MASS.

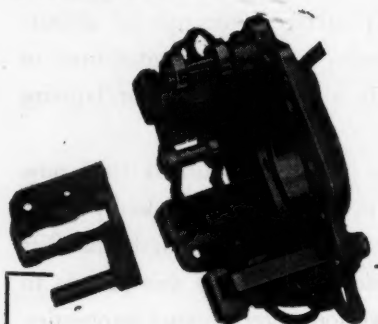
SUBSIDIARY OF INDUSTRIAL STEELS, INC.

DIE-CASTINGS
SIMPLIFY HOMELITE GENERATOR ASSEMBLIES

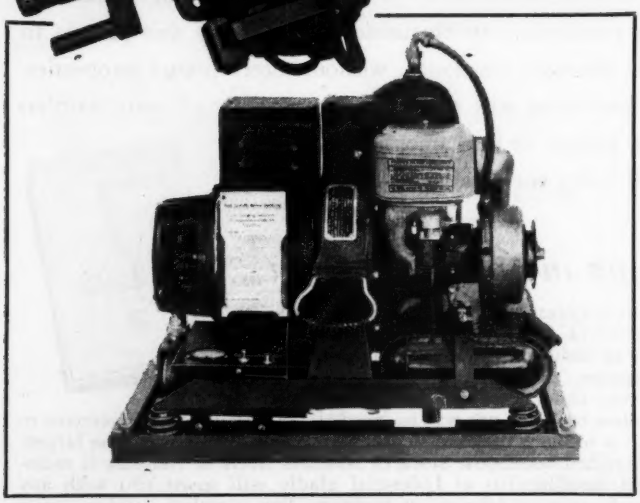
By die-casting brush holders and rings HOMELITE Corporation was able to junk a complicated rivet-and-weld assembly of twelve parts for a stronger, more dependable one of only two parts, in their well-known HOMELITE Portable Generators used in Army Liberators, Flying Forts and other planes.

In addition, by die-casting carburetor cover and fittings more time and money were saved.

Moreover, it was found that the tolerances in every case were close enough to make unnecessary any machining. Think what this means! With limited available manpower HOMELITE was able actually to turn out a better product faster and at lower cost.



This is no unique experience. Again and again, in all manner of industries, ADVANCE castings have made other forms of fabrication obsolete overnight. You, too, perhaps? Why not consult us? No obligation.



ADVANCE PRESSURE CASTINGS, INC.

ENGINEERS, DESIGNERS AND MANUFACTURERS OF DIE-CASTINGS OF "QUALITY WITH ECONOMY"

38 NORTH 15th STREET, BROOKLYN 22, NEW YORK

assistant general manager and chief engineer of the Gen May Corp., Logansport, Ind. Among other key positions those of Don Thomas who is connected with sales and engine ing, and Jack Marsh also in sales.

♦

Establishment of three new factory branches has been a nounced by Sterling Electric Motors Inc., Los Angeles. Al A. Adams has been appointed manager of the new Philadelp factory branch at 121 North Broad street. Mr. Adams h been a representative of the company in New York. Open the Detroit factory branch at 6432 Cass avenue will be L. H White, a factory-trained sales engineer. O. M. MacNaMa, a factory sales engineer, has been named manager of the ne Seattle factory branch at 920 Second avenue.

♦

In addition to his present position as vice president and ma ager of apparatus sales, C. H. Lang has also been given t direction of application and service engineering of Gen Electric Company's apparatus department. The appointm of A. K. Bushman as manager of application and service e gineering, apparatus department, has also been announced.

♦

Sales manager of the Fort Pitt Steel Casting Co. for t past ten years, Thomas F. Dorsey has joined Pittsburgh St Foundry Corp. of Glassport, Pa., and will serve as a spec representative.

♦

Appointment of Frank Roby as sales manager of the i dustrial controller division, Square D Co., with headquarters Milwaukee, has been announced. Mr. Roby, who left the co pany in 1942 for the armed services, replaces T. B. Martin w has assumed his new duties as director of advertising for t Detroit and Milwaukee electrical divisions.

♦

Recent changes in the sales department of PESCO Produc Co., Cleveland, include the promotion of E. S. Moreland general sales manager, A. E. Wilson as assistant sales manag for the automotive division, and Ray G. Holt and C. M. Smi as field representatives.

♦

Stackpole Carbon Co., St. Marys, Pa., has named Fra N. Townsend as export manager for the Electronic Compone Division, with offices at 254 West Thirty-first street, New Yor

♦

Associated with Formica Insulation Co. since 1923, Willi O. Miller has been assigned to the New York office, while Fra C. Konersman who has been with the engineering staff of t company for several years will assist Frank Manley, Detro office manager.

♦

In order to coordinate General Electric's diversified interes and afford better service, ten commercial vice presidents the company, representative of every sales district in the Unit States, will relinquish their responsibilities for apparatus sa and become members of the president's staff. Those affecte are: R. M. Alvord of San Francisco; T. F. Barton of New Yor W. O. Bathelder of Chicago; L. T. Blaisdell of Cleveland; W. Clayton of Dallas; E. H. Ginn of Atlanta; A. L. Jones of Denve T. S. Knight of Boston; A. S. Moody of Portland, Ore.; a C. K. West of Philadelphia. Taking over these duties w

Lengthen Service Life

...regardless of unit size!

↑ TIMES LONGER LIFE

ight years' service instead of o, was obtained by a large emical company through ecifying Nickel cast iron this 8-ton ring. Plain iron ed here corroded and oded ½" per year. This ickel alloyed iron wears s than ⅓" in the same e.

Increases Result/Dollar Ratio...

In products and applications . . . from a wrench to a railroad . . . records show that a little Nickel goes a long way to lengthen product life, to reduce bulk and deadweight, and to save maintenance costs.

When compared with costs involved, metals to which Nickel is added invariably rise sharply in service value . . . due to improved performance.

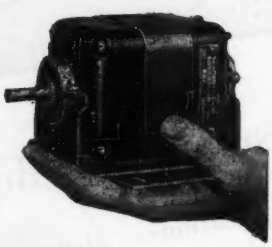
Nickel imparts strength, hardness, toughness, resistance to corrosion, heat and abrasive wear. In all fields of industry Nickel offers significant advantages. Send us details of your problem for our recommendations.

THE INTERNATIONAL NICKEL COMPANY, INC.

67 Wall Street, New York 5, N.Y.

OHIO
SMALL MOTORS

for General Purpose Applications



Types — Split Phase, Capacitor, Shaded Pole, and Direct Current
Speeds — 1100, 1700, 3400 RPM
Power 1/80 to 1/20 HP, depending on speed
Voltages—up to 440 A.C. and 250 D.C.
Frequencies — 60, 50, and 25 cycles
Bases—Solid, Resilient or Flange
Mountings—Horizontal, Sidewall, Ceiling or Vertical, with shaft up or down
Efficiency—High for such small powers

OHIO MOTORS INCLUDE

Split-phase induction to 1/3 HP	Torque to 100 oz. ft.
Direct Current to 1 HP	Shaded Pole to 1/30 HP
Capacitor to 2 HP	Synchronous to 1/4 HP
Polyphase to 2 HP	A.C. to D.C. Motor Generators to 500 watts
Aircraft Motors	High to Low Voltage D.C. Dynamotors up to 300 watts
Shell Type to 7½ HP	

What is your problem?

THE OHIO ELECTRIC MFG. CO.
5906 Maurice Avenue — Cleveland 4, Ohio

be: C. E. H. Palmer of New Haven, covering the New Eng[land] district with headquarters in Boston; W. J. Dorworth of Phi[la]delphia, the Atlantic district; C. L. Redd of Birmingham, [Ala.], Southeastern district with headquarters in Atlanta; C. W. Fi[sher] of Cleveland, the East Central district; R. I. Parker of Chicago, the Central district; H. A. White of Dallas, the Southwestern district; F. H. Doremus of Denver, the Rocky Mountain district; A. G. Jones of San Francisco, the Pacific district; and J. [E.] Murphy of Spokane, the Northwestern district, with headquarters in Portland. T. F. Barton of New York will continue for the present to act as the New York district manager in addition to his new duties on the president's staff.

♦

Purchase of the 30 Wythe Avenue building which adjoins present plant facilities has been announced by Advance Pressure Castings Inc. of Brooklyn, N. Y., manufacturer of die castings.

♦

H. L. Huntsinger has been appointed sales manager of the R-B-M Manufacturing Co., a division of Essex Wire Corp. Prior to his appointment he had been assistant to the sales manager of the Square D Co. at Milwaukee.

♦

Change of name has been announced by Crowe Name Plate & Mfg. Co., 3701 Ravenswood avenue, Chicago. Hereafter the company will be known as Croname Inc. Management, personnel and facilities will remain the same.

♦

Associated with the company for the past eight years, D. C. Green has been made district manager in the New York area for Haskelite Mfg. Corp., Grand Rapids, Mich. Previously Mr. Green had been district manager in the St. Louis area.

♦

Announcement has been made of the appointment of Joseph W. Wilson as manager of the Glendale, Calif., branch office for General Controls Co. Mr. Wilson will devote his time to serving users of automatic controls in the heating, refrigeration, aircraft and industrial fields throughout the Southern California, Arizona and Southern New Mexico area.

♦

Since 1940 manager of the Fiberglas Fabrics division in the Toledo general offices of Fiberglas Corp., Ben S. Wright has been transferred to Cleveland as manager of the corporation's branch office located at 825 Hanna building.

♦

Promotions of Howard Holmes to the position of sales manager and Russell Newcomb to sales promotion and advertising manager of Simmonds Aerocessories Inc., New York, has been announced.

♦

Two appointments have been made by United States Rubber Co.—that of H. S. McPherson of St. Louis as midwestern sales manager of the mechanical goods division and W. M. Ballew of Kansas City, Mo., as southwestern sales manager. Mr. McPherson's territory includes Detroit, Cincinnati, Indianapolis, Chicago, Milwaukee and St. Louis, while Mr. Ballew will be responsible for mechanical goods sales in branch offices in Kansas City, Tulsa, Denver, Houston, Dallas, New Orleans, Omaha and Minneapolis.

P & N DIRECT LINE

Swaged Hooks

OFFER NEW SAFETY FEATURES!

Sizes for 3/16" to 1" cable
Tightly swaged— Give foul-free surface
Straight shank
Precision forged of alloy steel
Smooth inside surface
Ample radius
Finished point
Load is carried on direct line from cable, prevents slipping

Now for the first time Poulsen & Nardon offer a complete line of hooks that can be swaged to cables from 3/16" to 1" in diameter. This permits faster, safer fastening. No more costly splicing with jagged, dangerous ends. P & N Direct Line Swaged Hooks are smooth, streamlined and foul-free!

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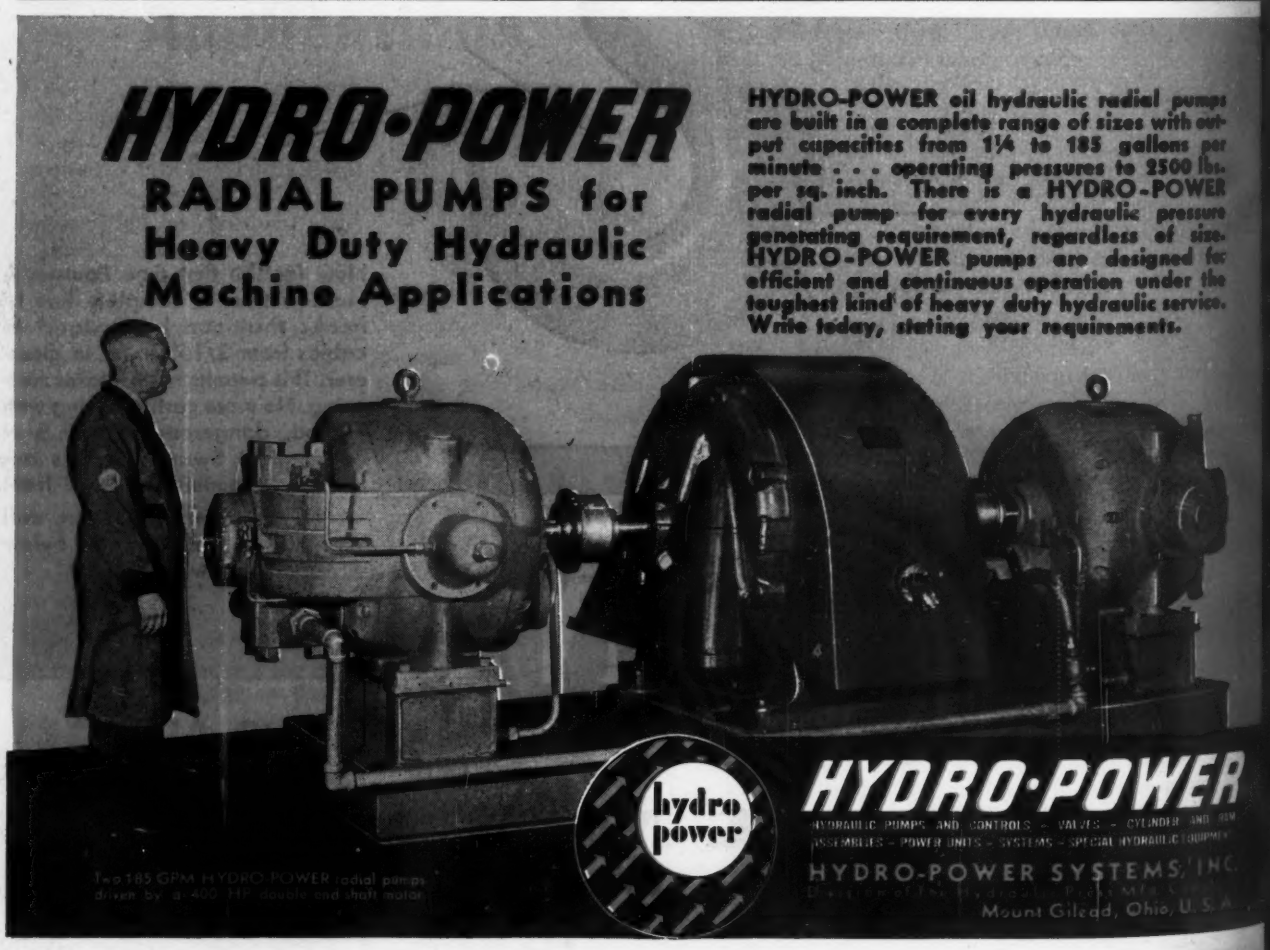
*Illustrated on Pages 160-163.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